

# TRANSACTIONS

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ENGINEERS

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*1894—Fifty Years—1944*

*A Great Past—A Greater Future*

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President.....T. F. Rockwell  
Vice-President.....G. G. Waters  
Secretary.....E. H. Riesmeyer, Jr.  
Treasurer.....L. S. Maehling  
Board of Governors: C. M. Humphreys, D. W. Loucks, A. F. Nass

### Rocky Mountain

Headquarters, Denver, Colo.  
Meets: *First Wednesday*

President.....J. H. McCabe  
Vice-President.....G. D. Maves  
Secretary.....F. L. Adams  
Treasurer.....W. M. Larimer  
Board of Governors: John Bernzen, J. J. Johnson,  
D. J. McQuaid

### St. Louis

Headquarters, St. Louis  
Meets: *First Tuesday*

President.....G. B. Rodenheiser,  
1st Vice-President.....W. J. Onk  
2nd Vice-President.....B. C. Simons  
Secretary.....B. L. Evans  
Treasurer.....L. R. Szombathy  
Board of Governors: C. F. Boester, B. L. Evans,  
E. A. Jones, J. S. Malone, W. J. Onk,  
G. B. Rodenheiser, B. C. Simons, L. R. Szombathy, Ralph Toensfeldt

# OFFICERS OF LOCAL CHAPTERS—1944 (continued)

## South Texas

Headquarters, Houston  
Meets: Third Friday

President.....A. B. Banowsky  
Vice-President.....A. F. Barnes  
Secretary.....B. F. Fisher  
Treasurer.....J. A. Walsh  
Board of Governors: A. M. Chase, C. A. McKinney, R. F. Taylor

## Washington, D. C.

Headquarters, Washington, D. C.  
Meets: Second Wednesday

President.....J. W. Markert  
Vice-President.....W. H. Littleford  
Secretary.....A. S. Gates, Jr.  
Treasurer.....M. F. Hoppe  
Board of Governors: I. M. Day, S. L. Gregg, Glegge Thomas

## Southern California

Headquarters, Los Angeles  
Meets: Second Wednesday

President.....Leo Hungerford  
Vice-President.....Maron Kennedy  
Secretary.....Art. Theobald  
Treasurer.....R. A. Lowe  
Board of Governors: J. G. Defton, F. B. Gardner, H. M. Hendrickson, W. O. Stewart

## Western Michigan

Headquarters, Grand Rapids  
Meets: Second Monday

President.....H. D. Bratt  
Vice-President.....H. J. Metzger  
Secretary.....Frank Harbin, Jr.  
Treasurer.....J. W. Miller  
Board of Governors: Arthur Boot, O. D. Marshall, C. H. Pesterfield

## Utah

Headquarters, Salt Lake City  
Meets: First Wednesday

President.....H. G. Richardson  
Vice-President.....J. T. Young, Jr.  
Secretary-Treasurer.....E. V. Gritton  
Board of Governors: R. C. Brown, D. B. Holford, C. E. Murdock, Alfred Richeda

## Western New York

Headquarters, Buffalo  
Meets: Second Monday

President.....S. W. Strouse  
1st Vice-President.....F. A. Moesel  
2nd Vice-President.....Herman Seelbach, Jr.  
Secretary.....G. E. Adema  
Board of Governors: M. C. Beman, Joseph Davis, W. R. Heath, D. J. Mahoney, S. M. Quackenbush

## Wisconsin

Headquarters, Milwaukee  
Meets: Third Monday

President.....I. J. Haus  
Vice-President.....O. A. Trostel  
Secretary.....M. W. Bishop  
Treasurer.....E. W. Gifford  
Board of Governors: T. M. Hughey, F. W. Goldsmith, C. H. Randolph

## Founding of the ASHVE

The organization of a Society of heating and ventilating engineers was first talked of by Hugh J. Barron and L. H. Hart in the early summer of 1894.

On August 2, 1894, a group of 15 men met at the office of *Heating and Ventilation*, 146 World Bldg., New York, and a committee of 5 was chosen to effect an organization.

The first meeting was called at 3:00 p.m., September 10, 1894, at the Broadway Central Hotel and temporary officers were elected: F. P. Smith, *Chairman*, and L. H. Hart, *Clerk*. The roll call showed that 75 persons had become Charter Members. It was voted that the name of the organization should be The American Society of Heating and Ventilating Engineers and a Constitution and By-Laws were adopted. Officers were elected to hold office until the 1st Annual Meeting, January 22-24, 1895.

### CHARTER MEMBERS OF THE SOCIETY

Henry Adams Washington, D. C.	C. F. Gessert New York, N. Y.	Charles W. Newton Baltimore, Md.
Homer Addams Washington, D. C.	Judson A. Goodrich New York, N. Y.	Theodore C. Northcott Elmira, N. Y.
Newell P. Andrus Brooklyn, N. Y.	John Gormly Philadelphia, Pa.	Charles S. Onderdonk Philadelphia, Pa.
Hugh J. Barron New York, N. Y.	James A. Harding New York, N. Y.	John A. Payne Providence, R. I.
Thomas Barwick New York, N. Y.	L. H. Hart New York, N. Y.	John H. Petherick Chattanooga, Tenn.
Edward P. Bates Syracuse, N. Y.	Charles F. Hausa New York, N. Y.	Geo. W. Plastow Jersey City, N. J.
Geo. C. Blackmore Pittsburgh, Pa.	Jno. D. Hibbard Chicago, Ill.	Henry B. Prather Buffalo, N. Y.
J. J. Blackmore New York, N. Y.	William H. Hill New York, N. Y.	Leon H. Prentice Chicago, Ill.
L. R. Blackmore New York, N. Y.	Geo. D. Hoffman Chicago, Ill.	D. M. Quay Chicago, Ill.
Samuel Burns New York, N. Y.	Charles S. Hopkins Rochester, N. Y.	Wm. A. Russell New York, N. Y.
B. Harold Carpenter Wilkes Barre, Pa.	Alfred A. Hunting Boston, Mass.	U. G. Scollay Brooklyn, N. Y.
R. C. Carpenter Ithaca, N. Y.	Stewart A. Jellett Philadelphia, Pa.	Percival H. Seward New York, N. Y.
Albert A. Cary New York, N. Y.	J. H. Kinealy St. Louis, Mo.	Le Roy B. Sherman New York, N. Y.
Robert C. Clarkson Philadelphia, Pa.	Jos. A. Langdon Pittsburgh, Pa.	Fred P. Smith New York, N. Y.
Geo. B. Cobb New York, N. Y.	Chas. W. Light Saginaw, Mich.	B. F. Stangland New York, N. Y.
H. D. Crane Cincinnati, Ohio	H. E. Light Saginaw, Mich.	Geo. P. Steel Philadelphia, Pa.
Albert A. Cryer New York, N. Y.	Chas. C. Lincoln New York, N. Y.	Joseph M. Stoughton Yonkers, N. Y.
T. B. Cryer Newark, N. J.	C. K. Longenecker New York, N. Y.	H. M. Swetland New York, N. Y.
Mark Dean Boston, Mass.	James Mackay Chicago, Ill.	Geo. H. Underhill Boston, Mass.
John Demarest Boston, Mass.	Wm. M. Mackay New York, N. Y.	T. J. Waters Chicago, Ill.
Thos. J. Douglass Norwich, Conn.	A. S. Mappett New York, N. Y.	J. R. Wendover New York, N. Y.
A. C. Edgar Philadelphia, Pa.	Wm. McMannis New York, N. Y.	W. B. Wilkinson New York, N. Y.
Hermann Eisert Baltimore, Md.	George L. Mehring Chicago, Ill.	James R. Willett Chicago, Ill.
John A. Fish Boston, Mass.	Edward A. Munro Brooklyn, N. Y.	J. J. Wilson Troy, N. Y.
Frank W. Foster Boston, Mass.	Robert Munro Pittsburgh, Pa.	Wiltzie F. Wolfe Boston, Mass.

# PAST OFFICERS

## AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

1894

*President*..... Edward P. Bates  
*1st Vice-President*..... Wm. M. Mackay  
*2nd Vice-President*..... Wiltzie F. Wolfe  
*3rd Vice-President*..... Chas. S. Onderdonk  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... L. H. Hart

### Board of Managers

*Chairman*, Fred P. Smith  
 Henry Adams A. A. Cary  
 Hugh J. Barron James A. Harding  
 Edward P. Bates, *Pres.* L. H. Hart, *Secy.*

### Council

*Chairman*, R. C. Carpenter  
 Albert A. Cryer Chas. W. Newton  
 F. W. Foster Ulysses G. Scollay, *Secy.*

1895

*President*..... Stewart A. Jellett  
*1st Vice-President*..... Wm. M. Mackay  
*2nd Vice-President*..... Chas. S. Onderdonk  
*3rd Vice-President*..... D. M. Quay  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... L. H. Hart

### Board of Managers

*Chairman*, James A. Harding  
 Geo. B. Cobb Ulysses G. Scollay  
 Wm. McMannis B. F. Stangland  
 Stewart A. Jellett, *Pres.* L. H. Hart, *Secy.*

### Council

*Chairman*, R. C. Carpenter  
 Henry Adams T. J. Waters  
 Edward P. Bates Albert A. Cryer, *Secy.*

1896

*President*..... R. C. Carpenter  
*1st Vice-President*..... D. M. Quay  
*2nd Vice-President*..... Edward P. Bates  
*3rd Vice-President*..... F. W. Foster  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... L. H. Hart

### Board of Managers

*Chairman*, Wm. M. Mackay  
 Hugh J. Barron Stewart A. Jellett  
 W. S. Hadaway, Jr. Wiltzie F. Wolfe  
 R. C. Carpenter, *Pres.* L. H. Hart, *Secy.*

### Council

*Chairman*, A. A. Cary  
 Albert A. Cryer B. F. Stangland  
 Wm. McMannis J. J. Blackmore, *Secy.*

1897

*President*..... Wm. M. Mackay  
*1st Vice-President*..... H. D. Crane  
*2nd Vice-President*..... Henry Adams  
*3rd Vice-President*..... A. E. Kenrick  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... H. M. Swetland

### Board of Managers

*Chairman*, R. C. Carpenter  
 Edward P. Bates Stewart A. Jellett  
 W. S. Hadaway, Jr. Wiltzie F. Wolfe  
 Wm. M. Mackay, *Pres.* H. M. Swetland, *Secy.*

### Council

*Chairman*, Albert A. Cryer  
 John A. Fish James Mackay  
 Wm. McMannis B. F. Stangland

1898

*President*..... Wiltzie F. Wolfe  
*1st Vice-President*..... J. H. Kinealy  
*2nd Vice-President*..... A. E. Kenrick  
*3rd Vice-President*..... John A. Fish  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... Stewart A. Jellett

### Board of Managers

*Chairman*, Wm. M. Mackay  
 Thomas Barwick A. C. Mott  
 John A. Connolly Francis A. Williams  
 Wiltzie F. Wolfe, *Pres.* Stewart A. Jellett, *Secy.*

### Council

*Chairman*, R. C. Carpenter  
 Henry Adams W. S. Hadaway, Jr.  
 Albert A. Cryer Wm. McMannis  
 Wiltzie F. Wolfe, *Pres.* Stewart A. Jellett, *Secy.*

1899

*President*..... Henry Adams  
*1st Vice-President*..... D. M. Quay  
*2nd Vice-President*..... A. E. Kenrick  
*3rd Vice-President*..... Francis A. Williams  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... Wm. M. Mackay

### Board of Managers

*Chairman*, Stewart A. Jellett  
 B. H. Carpenter Wm. Kent  
 A. A. Cary Wiltzie F. Wolfe  
 Henry Adams, *Pres.* Wm. M. Mackay, *Secy.*

### Council

*Chairman*, R. C. Carpenter  
 John Gormly Wm. McMannis  
 W. S. Hadaway, Jr. B. F. Stangland  
 Henry Adams, *Pres.* Wm. M. Mackay, *Secy.*

1900

*President*..... D. M. Quay  
*1st Vice-President*..... A. E. Kenrick  
*2nd Vice-President*..... Francis A. Williams  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... Wm. M. Mackay

### Board of Governors

*Chairman*, D. M. Quay  
 Wm. Kent, *Vice-Chm.* D. M. Nesbit  
 R. C. Carpenter C. B. J. Snyder  
 John Gormly Wm. M. Mackay, *Secy.*

1901

*President*..... J. H. Kinealy  
*1st Vice-President*..... A. E. Kenrick  
*2nd Vice-President*..... Andrew Harvey  
*Treasurer*..... Judson A. Goodrich  
*Secretary*..... Wm. M. Mackay

### Board of Governors

*Chairman*, J. H. Kinealy  
 Wm. Kent, *Vice-Chm.* John Gormly  
 R. C. Carpenter C. B. J. Snyder  
 R. P. Bolton Wm. M. Mackay, *Secy.*

# PAST OFFICERS (continued)

1902

President.....A. E. Kenrick  
1st Vice-President.....Andrew Harvey  
2nd Vice-President.....Robert C. Clarkson  
Treasurer.....Judson A. Goodrich  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, A. E. Kenrick  
John Gormly, Vice-Chm. J. H. Kinealy  
R. C. Carpenter C. B. J. Snyder  
Wm. Kent Wm. M. Mackay, Secy.

1903

President.....H. D. Crane  
1st Vice-President.....Wm. Kent  
2nd Vice-President.....R. P. Bolton  
Treasurer.....Judson A. Goodrich  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, H. D. Crane  
C. B. J. Snyder, Vice-Chm. A. E. Kenrick  
R. C. Carpenter Geo. Mehrling  
John Gormly Wm. M. Mackay, Secy.

1904

President.....Andrew Harvey  
1st Vice-President.....John Gormly  
2nd Vice-President.....Robert C. Clarkson  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, Andrew Harvey  
John Gormly H. D. Crane  
Robert C. Clarkson A. E. Kenrick  
J. J. Blackmore J. C. F. Snyder  
R. C. Carpenter Wm. M. Mackay, Secy.

1905

President.....Wm. Kent  
1st Vice-President.....R. P. Bolton  
2nd Vice-President.....C. B. J. Snyder  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, Wm. Kent  
R. P. Bolton James Mackay  
C. B. J. Snyder B. F. Stangland  
B. H. Carpenter J. C. F. Trachsel  
A. B. Franklin Wm. M. Mackay, Secy.

1906

President.....John Gormly  
1st Vice-President.....C. B. J. Snyder  
2nd Vice-President.....T. J. Waters  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, John Gormly  
C. B. J. Snyder, Vice-Chm. James Mackay  
R. C. Carpenter B. F. Stangland  
Frank K. Chew T. J. Waters  
A. B. Franklin Wm. M. Mackay, Secy.

1907

President.....C. B. J. Snyder  
1st Vice-President.....James Mackay  
2nd Vice-President.....Wm. G. Snow  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, C. B. J. Snyder  
James Mackay, Vice-Chm. Frank K. Chew  
R. E. Atkinson A. B. Franklin  
R. C. Carpenter Wm. G. Snow  
Edmund F. Capron Wm. M. Mackay, Secy.

1908

President.....James Mackay  
1st Vice-President.....Jas. D. Hoffman  
2nd Vice-President.....B. F. Stangland  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, James Mackay  
Jas. D. Hoffman, Vice-Chm. John F. Hale  
B. F. Stangland August Kehm  
R. C. Carpenter C. B. J. Snyder  
Frank K. Chew Wm. M. Mackay, Secy.

1909

President.....Wm. G. Snow  
1st Vice-President.....August Kehm  
2nd Vice-President.....B. S. Harrison  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, Wm. G. Snow  
August Kehm, Vice-Chm. Samuel R. Lewis  
John R. Allen James Mackay  
R. C. Carpenter B. F. Stangland  
B. S. Harrison Wm. M. Mackay, Secy.

1910

President.....Jas. D. Hoffman  
1st Vice-President.....R. P. Bolton  
2nd Vice-President.....Samuel R. Lewis  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. M. Mackay

## Board of Governors

Chairman, Jas. D. Hoffman  
R. P. Bolton, Vice-Chm. John F. Hale  
Geo. W. Barr Samuel R. Lewis  
R. C. Carpenter James Mackay  
Judson A. Goodrich Wm. M. Mackay, Secy.

1911

President.....R. P. Bolton  
1st Vice-President.....John R. Allen  
2nd Vice-President.....A. B. Franklin  
Treasurer.....Ulysses G. Scollay  
Secretary.....Wm. W. Macon

## Board of Governors

Chairman, R. P. Bolton  
John R. Allen, Vice-Chm. A. B. Franklin  
John T. Bradley Jas. D. Hoffman  
R. C. Carpenter August Kehm  
James H. Davis Wm. W. Macon, Secy.

# PAST OFFICERS (continued)

1912

President..... John R. Allen  
1st Vice-President..... John F. Hale  
2nd Vice-President..... Edmund F. Capron  
Treasurer..... James A. Donnelly  
Secretary..... Wm. W. Macon

## Board of Governors

Chairman, John R. Allen  
John F. Hale, Vice-Chm. Dwight D. Kimball  
Edmund F. Capron Samuel R. Lewis  
R. P. Bolton Wm. M. Mackay  
Jas. D. Hoffman Wm. W. Macon, Secy.

1913

President..... John F. Hale  
1st Vice-President..... A. B. Franklin  
2nd Vice-President..... Edmund F. Capron  
Treasurer..... James A. Donnelly  
Secretary..... Edwin A. Scott

## Board of Governors

Chairman, John F. Hale  
A. B. Franklin, Vice-Chm. James A. Donnelly  
John R. Allen Dwight D. Kimball  
Edmund F. Capron Wm. W. Macon  
R. P. Bolton James M. Stannard  
Frank T. Chapman Theodore Weinschank  
Ralph Collamore Edwin A. Scott, Secy.

1914

President..... Samuel R. Lewis  
1st Vice-President..... Edmund F. Capron  
2nd Vice-President..... Dwight D. Kimball  
Treasurer..... James A. Donnelly  
Secretary..... J. J. Blackmore

## Council

Chairman, Samuel R. Lewis  
E. F. Capron, Vice-Chm. John F. Hale  
Dwight D. Kimball Harry M. Hart  
John R. Allen Frank G. McCann  
Frank T. Chapman Wm. W. Macon  
Frank I. Cooper James M. Stannard  
James A. Donnelly J. J. Blackmore, Secy.

1915

President..... Dwight D. Kimball  
1st Vice-President..... Harry M. Hart  
2nd Vice-President..... Frank T. Chapman  
Treasurer..... Homer Addams  
Secretary..... J. J. Blackmore

## Council

Chairman, Dwight D. Kimball  
Harry M. Hart, Vice-Chm. Samuel R. Lewis  
Homer Addams Frank G. McCann  
Frank T. Chapman J. T. J. Mellon  
Frank I. Cooper Henry C. Meyer, Jr.  
E. Vernon Hill Arthur K. Ohmes  
Wm. M. "Ingsbury J. J. Blackmore, Secy.

1916

President..... Harry M. Hart  
1st Vice-President..... Frank T. Chapman  
2nd Vice-President..... Arthur K. Ohmes  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, Harry M. Hart  
F. T. Chapman, Vice-Chm. Dwight D. Kimball  
Homer Addams Henry C. Meyer, Jr.  
Charles R. Blahop Arthur K. Ohmes  
Frank I. Cooper Fred R. Still  
Milton W. Franklin Walter S. Timmis  
E. Vernon Hill Casin W. Obert, Secy.

1917

President..... J. Irvine Lyle  
1st Vice-President..... Arthur K. Ohmes  
2nd Vice-President..... Fred R. Still  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, J. Irvine Lyle  
A. K. Ohmes, Vice-Chm. Harry M. Hart  
Homer Addams E. Vernon Hill  
Davis S. Boyden James M. Stannard  
Bert C. Davis Fred R. Still  
Milton W. Franklin Walter S. Timmis  
Charles A. Fuller Casin W. Obert, Secy.

1918

President..... Fred R. Still  
1st Vice-President..... Walter S. Timmis  
2nd Vice-President..... E. Vernon Hill  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, Fred R. Still  
W. S. Timmis, Vice-Chm. J. Irvine Lyle  
Homer Addams E. Vernon Hill  
William H. Driscoll Frank G. Phegley  
Howard H. Fielding Fred. W. Powers  
H. P. Gant Champlain L. Riley  
C. W. Kimball Casin W. Obert, Secy.

1919

President..... Walter S. Timmis  
1st Vice-President..... E. Vernon Hill  
2nd Vice-President..... Milton W. Franklin  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, Walter S. Timmis  
E. Vernon Hill, Vice-Chm. Frank G. Phegley  
Homer Addams Fred. W. Powers  
Howard H. Fielding Robt. W. Pryor, Jr.  
Milton W. Franklin Champlain L. Riley  
Harry E. Gerriah Fred R. Still  
George B. Nichols Casin W. Obert, Secy.

1920

President..... E. Vernon Hill  
1st Vice-President..... Champlain L. Riley  
2nd Vice-President..... Jay R. McColl  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, E. Vernon Hill  
C. L. Riley, Vice-Chm. Jay R. McColl  
Homer Addams George B. Nichols  
Jos. A. Cutler Robt. W. Pryor, Jr.  
Wm. H. Driscoll W. S. Timmis  
A. C. Edgar Perry West  
Alfred Kellogg Casin W. Obert, Secy.

1921

President..... Champlain L. Riley  
1st Vice-President..... Jay R. McColl  
2nd Vice-President..... H. P. Gant  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, Champlain L. Riley  
Jay R. McColl, Vice-Chm. E. S. Hallett  
Homer Addams E. Vernon Hill  
Jos. A. Cutler Alfred Kellogg  
Samuel E. Dibble E. E. McNair  
Wm. H. Driscoll Perry West  
H. P. Gant Casin W. Obert, Secy.

# PAST OFFICERS (continued)

1922

President..... Jay R. McColl  
1st Vice-President..... H. P. Gant  
2nd Vice-President..... Samuel E. Dibble  
Treasurer..... Homer Addams  
Secretary..... Casin W. Obert

## Council

Chairman, Jay R. McColl  
H. P. Gant, Vice-Chm. L. A. Harding  
Homer Addams E. E. McNair  
Jos. A. Cutler H. J. Meyer  
Samuel E. Dibble C. L. Riley  
Wm. H. Driscoll Perry West  
E. S. Hallett Casin W. Obert, Secy.

1923

President..... H. P. Gant  
1st Vice-President..... Homer Addams  
2nd Vice-President..... E. E. McNair  
Treasurer..... Wm. H. Driscoll  
Secretary..... Casin W. Obert

## Council

Chairman, H. P. Gant  
Homer Addams, Vice-Chm. E. S. Hallett  
W. H. Carrier Alfred Kellogg  
J. A. Cutler Thornton Lewis  
S. E. Dibble E. E. McNair  
Wm. H. Driscoll Perry West  
Casin W. Obert, Secy.

1924

President..... Homer Addams  
1st Vice-President..... S. E. Dibble  
2nd Vice-President..... William H. Driscoll  
Treasurer..... Perry West  
Secretary..... F. C. Houghten

## Council

Chairman, Homer Addams  
S. E. Dibble, Vice-Chm. W. E. Gillham  
F. Paul Anderson L. A. Harding  
W. H. Carrier Alfred Kellogg  
J. A. Cutler Thornton Lewis  
William H. Driscoll Perry West  
H. P. Gant F. C. Houghten, Secy.

1925

President..... S. E. Dibble  
1st Vice-President..... Wm. H. Driscoll  
2nd Vice-President..... F. Paul Anderson  
Treasurer..... Perry West  
Secretary..... F. C. Houghten

## Council

Chairman, S. E. Dibble  
Wm. H. Driscoll, Vice-Chm. W. T. Jones  
Homer Addams Thornton Lewis  
F. Paul Anderson J. H. Walker  
W. H. Carrier Perry West  
J. A. Cutler A. C. Willard  
W. E. Gillham F. C. Houghten, Secy.

1926

President..... W. H. Driscoll  
1st Vice-President..... F. Paul Anderson  
2nd Vice-President..... A. C. Willard  
Treasurer..... W. E. Gillham  
Secretary..... A. V. Hutchinson

## Council

Chairman, W. H. Driscoll  
F. Paul Anderson, Vice-Chm. C. V. Haynes  
W. H. Carrier W. T. Jones  
J. A. Cutler E. B. Langenberg  
S. E. Dibble Thornton Lewis  
W. E. Gillham J. F. McIntire

A. C. Willard

1927

President..... F. Paul Anderson  
1st Vice-President..... A. C. Willard  
2nd Vice-President..... Thornton Lewis  
Treasurer..... W. E. Gillham  
Secretary..... A. V. Hutchinson

## Council

Chairman, F. Paul Anderson  
A. C. Willard, Vice-Chm. John Howatt  
H. H. Angus W. T. Jones  
W. H. Carrier J. J. Kissick  
W. H. Driscoll E. B. Langenberg  
Roswell Farnham Thornton Lewis  
H. H. Fielding J. F. McIntire  
W. E. Gillham H. Lee Moore  
C. V. Haynes F. B. Rowley

1928

President..... A. C. Willard  
1st Vice-President..... Thornton Lewis  
2nd Vice-President..... L. A. Harding  
Treasurer..... W. E. Gillham  
Secretary..... A. V. Hutchinson

## Council

Chairman, A. C. Willard  
Thornton Lewis, Vice-Chm. C. V. Haynes  
F. Paul Anderson John Howatt  
H. H. Angus W. T. Jones  
W. H. Carrier J. J. Kissick  
N. W. Downes E. B. Langenberg  
Roswell Farnham J. F. McIntire  
W. E. Gillham H. Lee Moore

F. B. Rowley

1929

President..... Thornton Lewis  
1st Vice-President..... L. A. Harding  
2nd Vice-President..... W. H. Carrier  
Treasurer..... W. E. Gillham  
Secretary..... A. V. Hutchinson  
Technical Secretary..... P. D. Close

## Council

Chairman, Thornton Lewis  
L. A. Harding, Vice-Chm. John Howatt  
H. H. Angus W. T. Jones  
W. H. Carrier E. B. Langenberg  
N. W. Downes G. L. Larson  
Roswell Farnham F. C. McIntosh  
W. E. Gillham W. A. Rowe  
C. V. Haynes F. B. Rowley

A. C. Willard

1930

President..... L. A. Harding  
1st Vice-President..... W. H. Carrier  
2nd Vice-President..... F. B. Rowley  
Treasurer..... C. W. Farrar  
Secretary..... A. V. Hutchinson  
Technical Secretary..... P. D. Close

## Council

Chairman, L. A. Harding  
W. H. Carrier, Vice-Chm. John Howatt  
H. H. Angus W. T. Jones  
D. S. Boyden E. B. Langenberg  
R. H. Carpenter G. L. Larson  
J. D. Cassell Thornton Lewis  
N. W. Downes F. C. McIntosh  
Roswell Farnham W. A. Rowe  
C. W. Farrar F. B. Rowley

# PAST OFFICERS (continued)

1931  
*President*..... W. H. Carrier  
*1st Vice-President*..... F. B. Rowley  
*2nd Vice-President*..... W. T. Jones  
*Treasurer*..... F. D. Mensing  
*Secretary*..... A. V. Hutchinson  
*Technical Secretary*..... P. D. Close

**Council**  
*Chairman, W. H. Carrier*  
 F. B. Rowley, *Vice-Chm.*  
 D. S. Boyden  
 E. K. Campbell  
 R. H. Carpenter  
 J. D. Cassell  
 E. O. Eastwood  
 Roswell Farnham  
 E. H. Gurney  
 L. A. Harding  
 John Howatt  
 W. T. Jones  
 E. B. Langenberg  
 G. L. Larson  
 F. C. McIntosh  
 F. D. Mensing  
 W. A. Rowe

1932  
*President*..... F. B. Rowley  
*1st Vice-President*..... W. T. Jones  
*2nd Vice-President*..... C. V. Haynes  
*Treasurer*..... F. D. Mensing  
*Secretary*..... A. V. Hutchinson  
*Technical Secretary*..... P. D. Close

**Council**  
*Chairman, F. B. Rowley*  
 W. T. Jones, *Vice-Chm.*  
 D. S. Boyden  
 E. K. Campbell  
 R. H. Carpenter  
 W. H. Carrier  
 John D. Cassell  
 E. O. Eastwood  
 Roswell Farnham  
 F. E. Giesecke  
 E. H. Gurney  
 C. V. Haynes  
 John Howatt  
 G. L. Larson  
 J. F. McIntire  
 F. D. Mensing  
 W. E. Stark

1933  
*President*..... W. T. Jones  
*1st Vice-President*..... C. V. Haynes  
*2nd Vice-President*..... John Howatt  
*Treasurer*..... D. S. Boyden  
*Secretary*..... A. V. Hutchinson

**Council**  
*Chairman, W. T. Jones*  
 C. V. Haynes, *Vice-Chm.*  
 D. S. Boyden  
 E. K. Campbell  
 R. H. Carpenter  
 J. D. Cassell  
 E. O. Eastwood  
 Roswell Farnham  
 F. E. Giesecke  
 E. H. Gurney  
 John Howatt  
 G. L. Larson  
 J. F. McIntire  
 F. C. McIntosh  
 L. W. Moon  
 F. B. Rowley  
 W. E. Stark

1934  
*President*..... C. V. Haynes  
*1st Vice-President*..... John Howatt  
*2nd Vice-President*..... G. L. Larson  
*Treasurer*..... D. S. Boyden  
*Secretary*..... A. V. Hutchinson

**Council**  
*Chairman, C. V. Haynes*  
 John Howatt, *Vice-Chm.*  
 M. C. Beman  
 D. S. Boyden  
 Albert Buenger  
 R. H. Carpenter  
 J. D. Cassell  
 F. E. Giesecke  
 E. H. Gurney  
 W. T. Jones  
 G. L. Larson  
 J. F. McIntire  
 F. C. McIntosh  
 L. Walter Moon  
 O. W. Ott  
 W. A. Russell  
 W. E. Stark

1935  
*President*..... John Howatt  
*1st Vice-President*..... G. L. Larson  
*2nd Vice-President*..... D. S. Boyden  
*Treasurer*..... A. J. Offner  
*Secretary*..... A. V. Hutchinson

**Council**  
*Chairman, John Howatt*  
 G. L. Larson, *Vice-Chm.*  
 M. C. Beman  
 D. S. Boyden  
 Albert Buenger  
 R. H. Carpenter  
 J. D. Cassell  
 F. E. Giesecke  
 E. H. Gurney  
 C. V. Haynes  
 J. F. McIntire  
 F. C. McIntosh  
 L. Walter Moon  
 A. J. Offner  
 O. W. Ott  
 W. A. Russell  
 W. E. Stark

1936  
*President*..... G. L. Larson  
*1st Vice-President*..... D. S. Boyden  
*2nd Vice-President*..... E. H. Gurney  
*Treasurer*..... A. J. Offner  
*Secretary*..... A. V. Hutchinson

**Council**  
*Chairman, G. L. Larson*  
 D. S. Boyden, *Vice-Chm.*  
 M. C. Beman  
 R. C. Bolsinger  
 Albert Buenger  
 S. H. Downs  
 W. L. Fleisher  
 F. E. Giesecke  
 E. H. Gurney  
 John Howatt  
 C. M. Humphreys  
 L. Walter Moon  
 J. F. McIntire  
 A. J. Offner  
 O. W. Ott  
 W. A. Russell  
 W. E. Stark

1937  
*President*..... D. S. Boyden  
*1st Vice-President*..... E. Holt Gurney  
*2nd Vice-President*..... J. F. McIntire  
*Treasurer*..... A. J. Offner  
*Secretary*..... A. V. Hutchinson

**Council**  
*Chairman, D. S. Boyden*  
 E. H. Gurney, *Vice-Chm.*  
 J. J. Aeberly  
 M. C. Beman  
 R. C. Bolsinger  
 Albert Buenger  
 S. H. Downs  
 E. O. Eastwood  
 W. L. Fleisher  
 F. E. Giesecke  
 C. M. Humphreys  
 G. L. Larson  
 J. F. McIntire  
 A. J. Offner  
 W. A. Russell

1938  
*President*..... E. Holt Gurney  
*1st Vice-President*..... J. F. McIntire  
*2nd Vice-President*..... F. E. Giesecke  
*Treasurer*..... A. J. Offner  
*Secretary*..... A. V. Hutchinson  
*Technical Secretary*..... John James

**Council**  
*Chairman, E. Holt Gurney*  
 J. F. McIntire, *Vice-Chm.*  
 N. D. Adams  
 J. J. Aeberly  
 M. C. Beman  
 R. C. Bolsinger  
 D. S. Boyden  
 S. H. Downs  
 E. O. Eastwood  
 W. L. Fleisher  
 F. E. Giesecke  
 C. M. Humphreys  
 A. P. Kratz  
 A. J. Offner  
 W. A. Russell  
 J. H. Walker  
 G. L. Wiggs

1939  
*President*..... J. F. McIntire  
*1st Vice-President*..... F. E. Giesecke  
*2nd Vice-President*..... W. L. Fleisher  
*Treasurer*..... M. F. Blankin  
*Secretary*..... A. V. Hutchinson  
*Technical Secretary*..... John James

**Council**  
*Chairman, J. F. McIntire*  
 F. E. Giesecke, *Vice-Chm.*  
 N. D. Adams  
 J. J. Aeberly  
 M. C. Beman  
 M. F. Blankin  
 E. K. Campbell  
 S. H. Downs  
 E. O. Eastwood  
 W. L. Fleisher  
 E. H. Gurney  
 A. P. Kratz  
 A. J. Offner  
 W. A. Russell  
 G. L. Tuve  
 J. H. Walker  
 G. L. Wiggs

# PAST OFFICERS (continued)

1940  
 President..... F. E. Giesecke  
 1st Vice-President..... W. L. Fleisher  
 2nd Vice-President..... E. O. Eastwood  
 Treasurer..... M. F. Blankin  
 Secretary..... A. V. Hutchinson  
 Technical Secretary..... John James

## Council

Chairman, F. E. Giesecke  
 W. L. Fleisher, Vice-Chm. E. N. McDonnell  
 N. D. Adams J. F. McIntire  
 M. F. Blankin A. J. Offner  
 E. K. Campbell G. L. Tuve  
 J. F. S. Collins, Jr. T. H. Urdahl  
 S. H. Downs J. H. Walker  
 E. O. Eastwood G. L. Wiggs  
 A. P. Kratz C.-E. A. Winslow

1942  
 President..... E. O. Eastwood  
 1st Vice-President..... M. F. Blankin  
 2nd Vice-President..... S. H. Downs  
 Treasurer..... E. K. Campbell  
 Secretary..... A. V. Hutchinson  
 Technical Secretary..... John James

## Council

Chairman, E. O. Eastwood  
 M. F. Blankin, Vice-Chm. A. J. Offner  
 E. K. Campbell W. A. Russell  
 J. F. Collins, Jr. L. P. Saunders  
 S. H. Downs A. E. Stacey, Jr.  
 W. L. Fleisher C. Tasker  
 A. P. Kratz T. H. Urdahl  
 E. N. McDonnell C.-E. A. Winslow  
 L. G. Miller B. M. Woods  
 F. C. McIntosh, Ex-Officio

1941  
 President..... W. L. Fleisher  
 1st Vice-President..... E. O. Eastwood  
 2nd Vice-President..... J. H. Walker  
 Treasurer..... M. F. Blankin  
 Secretary..... A. V. Hutchinson  
 Technical Secretary..... John James

## Council

Chairman, W. L. Fleisher  
 E. O. Eastwood, Vice-Chm. A. J. Offner  
 M. F. Blankin W. A. Russell  
 E. K. Campbell L. P. Saunders  
 J. F. Collins, Jr. C. Tasker  
 S. H. Downs G. L. Tuve  
 F. E. Giesecke T. H. Urdahl  
 A. P. Kratz J. H. Walker  
 E. N. McDonnell C.-E. A. Winslow  
 A. E. Stacey, Jr., Ex-Officio

1943  
 President..... M. F. Blankin  
 1st Vice-President..... S. H. Downs  
 2nd Vice-President..... C.-E. A. Winslow  
 Treasurer..... E. K. Campbell  
 Secretary..... A. V. Hutchinson  
 Technical Secretary..... Carl H. Flink

## Council

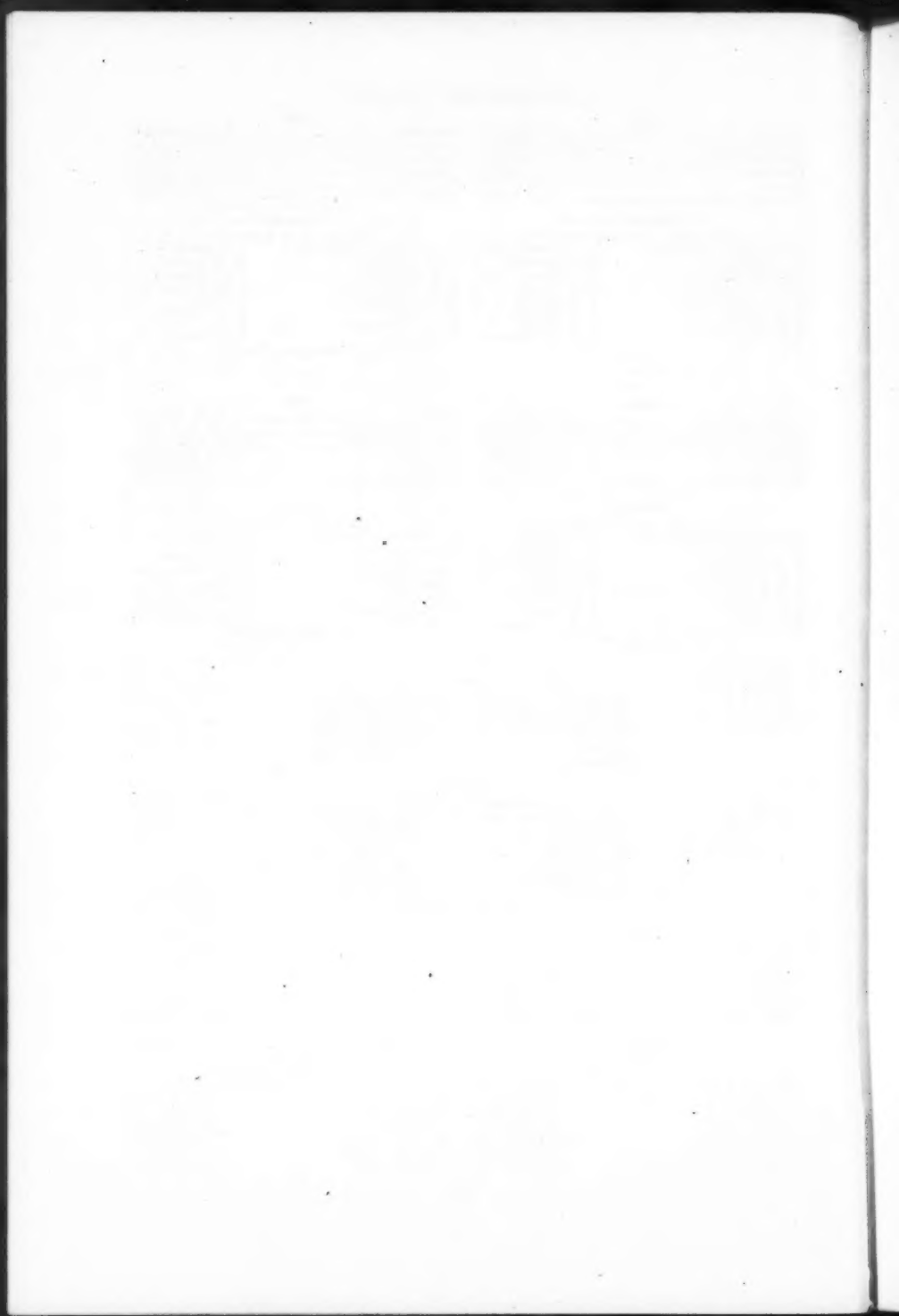
Chairman, M. F. Blankin  
 S. H. Downs, Vice-Chm. A. J. Offner  
 E. K. Campbell W. A. Russell  
 J. F. Collins, Jr. L. P. Saunders  
 E. O. Eastwood A. E. Stacey, Jr.  
 James Holt C. Tasker  
 A. P. Kratz T. H. Urdahl  
 E. N. McDonnell C.-E. A. Winslow  
 L. G. Miller B. M. Woods  
 C. M. Ashley, Ex-Officio

## 1944

President..... S. H. Downs  
 1st Vice-President..... C.-E. A. Winslow  
 2nd Vice-President..... Alfred J. Offner  
 Treasurer..... L. P. Saunders  
 Secretary..... A. V. Hutchinson  
 Technical Secretary..... Carl H. Flink

## Council

Chairman, S. H. Downs  
 C.-E. A. Winslow, Vice-Chm. E. N. McDonnell  
 C. M. Ashley L. G. Miller  
 L. T. Avery L. E. Seeley  
 M. F. Blankin A. E. Stacey, Jr.  
 J. F. Collins, Jr. T. H. Urdahl  
 W. A. Danielson G. D. Winans  
 James Holt B. M. Woods  
 G. L. Tuve, Ex-Officio



# TRANSACTIONS

of

## AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1243

### FIFTIETH ANNUAL MEETING, 1944 New York, N. Y.

THE 50th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in New York, N. Y., January 30-31, February 1-2, 1944 established a record by a total registration of 920, of which 540 were Society members, 258 guests and 122 ladies. Nineteen of the 23 Past Presidents were in attendance.

#### FIRST SESSION—MONDAY, JANUARY 31, 9:30 A.M.

The first session in Hotel Pennsylvania was called to order by Pres. M. F. Blankin, who stated that the greetings of the Honorable F. H. LaGuardia, Mayor of the City of New York, would be presented at the get-together luncheon.

Congratulatory messages were presented from various organizations (see p. 65).

President Blankin called upon Alfred J. Offner, *General Chairman* of the Committee on Arrangements for the 50th Annual Meeting, who extended a cordial welcome on behalf of New York Chapter.

President Blankin then announced the appointment of the Resolutions Committee consisting of R. A. Folsom, San Francisco, *Chairman*, H. Berkley Hedges, Philadelphia, and William Glass, Winnipeg, Canada.

#### ELECTION OF HONORARY MEMBERS

President Blankin then stated that due to the death of Reginald Pelham Bolton there were no living Honorary Members of the Society. He announced that the Council was unanimous in endorsing the petitions nominating Dr. Willis H. Carrier, Syracuse, N. Y., and Dr. Arno C. Fieldner, Washington, D. C., for election as Honorary Members of the Society. He briefly reviewed the distinguished careers of the candidates.

On motion of Alfred J. Offner, seconded by T. T. Tucker, Atlanta, it was unanimously voted that Dr. Willis H. Carrier be elected an Honorary Member of the Society.

On motion of Thornton Lewis, Washington, D. C., seconded by W. A. Russell, Kansas City, Mo., it was voted that Dr. Arno C. Fieldner be elected an Honorary Member of the Society.

President Blankin then submitted his report on the present condition of the Society as follows:

#### REPORT OF PRESIDENT

The 50th Annual Meeting of the Society marks an outstanding milestone in the life of a great, scientific organization.

In the midst of the greatest war of all time we are met here in New York these three days to celebrate our glorious history, consider the present, and prepare for the future.

Since our 49th Annual Meeting in Cincinnati, much has transpired that will provide for that future. Our financial position has improved. We have enlarged the scope of our research work and our technical activities. Our membership has been increased and is at a new all-time high. Our chapters have been strengthened. We have added a new chapter and are planning others. Now let me enlarge a little on some of these statements.

*First—Our Financial Position:* We recently changed our financial year so that it ended October 31. In order to get on this new basis it was necessary for last year to only cover the ten month period from January 1 to October 31, with our new year beginning November 1, 1943, extending to October 31, 1944.

For the first time within my knowledge, we have brought our Reserve Fund up to the requirements of the By-Laws, and that is—\$15.00 for each member. We now have in this fund \$54,395.45 of which \$52,241.82 is in Government Securities, and the balance in cash and savings funds. We have subscribed heavily to the various War Loans. In 1942 we purchased \$54,000.00 worth at par value; in 1943 we purchased \$18,000.00 worth at par value and \$8000.00 worth of Canadian Fourth and Fifth Victory Loan Bonds. The par value of all our U. S. Government and Canadian Bonds totals \$117,000.00.

The Finance Committee was quite pessimistic about our income this year due to resignations, members in the Service, and possible curtailment of advertising and sales of THE GUIDE. However, with the influx of new members and increased advertising and Guide sales, we were able to keep up the standards of services to our members and still able to show a slight excess of income in our budget.

*Second—Research and Technical Activities:* As we started this year, we were without a Technical Secretary due to the resignation of John James, and as a carry-over from the preceding year we had no Director of the Laboratory due to the call of Lieut.-Comdr. F. C. Houghten by the Navy. After a thorough discussion it was decided that now was the time to enlarge the scope of our research work and technical activities. This was to be accomplished by the employing of a new Technical Secretary who would concentrate his work on THE GUIDE, and Codes and Standards, not devoting half his time to the Research Committee as was previously done. We were fortunate in securing Carl H. Flink for this position, which he has filled in a very creditable manner. Methods of handling Codes and Standards have been set up under the direction of L. P. Saunders, chairman of this committee, and I am sure that this troublesome problem, in the future, will function quite smoothly and new Codes and Standards will be evolved as rapidly as they may be needed.

It was also considered advisable to create a new position, that of the Director of Research. This was done by Council and after a very diligent search we were quite fortunate in finding an outstanding man, who brought great honor to the Society in filling this position. I refer to Cyril Tasker from the Ontario Research Foundation, Ontario, Canada, who took over this office on October 1, 1943.

It was also brought to the attention of the Council that it was desirable to move the Laboratory from the Bureau of Mines in Pittsburgh, after 25 years of very pleasant association. We needed more room and the Bureau of Mines needed some of the space that we were occupying, so a committee was appointed at the Fall meeting of the Council with instructions to investigate the various proposals and suggestions made to the Society regarding the re-location of its Research Laboratory; select a new location for the Laboratory and make any arrangements that were

appropriate for the headquarters of the Society. This Committee consisted of C. M. Ashley, Chairman of the Research Committee; J. F. Collins, Jr., Chairman of the Finance Committee; E. N. McDonnell, a member of the Executive Committee; and the President as Chairman.

After investigating twelve proposals from various Universities and Institutions, the Committee unanimously selected the former Western Reserve Historical Society Building located at the corner of Euclid Avenue and East 107th St., Cleveland, Ohio. Our new Research Laboratory Building which we will occupy for at least the next two years will require the entire building, not just a small portion of it. The building is adjacent to the campuses of both Case School of Applied Science and Western Reserve University.

The Committee has further recommended that for the present the headquarters of the Society be maintained at its present location in New York, but that a committee be immediately appointed by Council to further investigate the possibility of combining the Laboratory and headquarters in one building so that such a move can be made immediately after the war, or before, if possible. I am sure that with our greater facilities, our enlarged scope of activities and the extension of our cooperative work with Universities and Technical Schools, it will be possible to attract a whole lot more interest on the part of industry in the great research work we are doing, and make even more feasible the possibility of our combined Laboratory and headquarters building.

*Third—Membership:* I am quite pleased with the splendid increase in membership this year, in spite of predictions that we would suffer a still further loss in members. In 1942 we had a loss of 380 members due to cancellations, resignations, death, and the creation of Life Members. We took in only 217 new members which gave us a net loss of 163.

Starting the year we had a total of 3006 members which, compared with 3147 members as of January 1, 1940, indicated that we were going backward, and a great Society like ours cannot afford to stand still on membership, let alone go backward.

We set a goal of 300 new members this year. How well that effort succeeded can be shown by the fact that we have actually taken in during the year 404 new members, and have on file applications from 23 potential members. Due to illness in the headquarters office during the last two months it has been impossible to process these through in the usual orderly manner, which means about 60 days. If all these are passed, our total membership in spite of deaths, resignations and cancellations, will then be over 3300 which is a new all-time high and an additional reason for celebrating our Golden Anniversaries.

It seems to me that this Society should have at least 5000 members, as there are many potential members who have probably never been approached.

I was quite interested in reading Pres. F. R. Still's Report at the 25th Annual Meeting here in New York, at which time we had 854 Members. He commented on the need for greatly increasing the membership and appealed for a membership of 1500. The members responded by adding 273 new members in the first four months. I happened to be one of them (1230 January 1, 1920).

Fifteen years later, again in New York, Pres. W. T. Jones commented on the membership and while we then had 1900 members, he felt that there were at least 2000 men who were well qualified for membership who did not belong to the Society.

If he could visualize a membership of 4000 at that time, certainly with the tremendous increase in our profession and industry in the last ten years a potential membership of 5000 is by no means beyond the realm of possibility.

Let's work with the thought in mind that this can be accomplished in the next three years.

*Fourth—Chapters:* I had hoped this year to be able to visit all the chapters, but unfortunately, due to conflicting dates, did not quite make all of them, despite the fact that I traveled nearly 30,000 miles, which is quite a bit of traveling under wartime conditions. Mrs. Blankin accompanied me for over 20,000 miles and again we wish to express our sincere thanks to all the chapter officers and members and their wives for their most gracious hospitality. I only wish that every member of the Society could make the same trip and actually get to know the individual members

as you meet them under these circumstances. They simply devote their entire time to you during your stay in their city and do everything possible to make it most pleasant and entertaining. Most of the chapters were functioning in excellent order, holding regular meetings that were well attended. A very few felt that because of war work, rationing, and loss of members, due to temporary change of working location, as well as induction into the Armed Services, interest could not be maintained, and had suspended most of their meetings.

We held an excellent meeting in each case, at which time it was pointed out to them the necessity for continuing to hold regular meetings to keep their chapter organization together, and furthermore that they were overlooking the possibility of securing new members under wartime conditions. Many were seeking admission but were waiting for someone to ask them to join.

On checking back I was pleased to learn that these chapters were holding regular meetings, and as a result members from that area have proposed quite a few new members into the Society. One meeting that I attended of a chapter that had suspended operations, resulted in eight applications that night.

In addition to visiting the chapters, I held meetings with our membership in Baltimore, Denver, San Antonio, and Indianapolis, with the hope that new chapters might be conceived.

At Denver we had a meeting under the auspices of the Colorado Section of the *ASME* with a large attendance and 47 potential members of our Society in that group. I had hoped to have a new chapter there before this, but unfortunately to date it has not materialized.

We should have chapters in Baltimore and San Antonio, and I am sure that in due time this will come about.

With a chapter in Baltimore we would probably increase our membership by at least 100. In fact, even more than that if we use what happened in Indianapolis as a guide.

A meeting was held with our members in Indianapolis early in September, with Life Member, Joseph G. Hayes, acting as Chairman. As a result of that meeting a chapter was organized and officers were elected, with an application for a charter for the Indiana Chapter signed by 24 members. This was granted by the Council early in October, and November 1 set as the date for the official presentation of the charter, and the Fall Council Meeting was set for October 31 in Indianapolis.

Then a most inspiring thing happened—the members in Indiana started soliciting new members. Two days before the meeting I had a letter from one of their officers and he was quite proud of the fact that they had 27 applications for membership and 2 applications for re-instatement. But that was only a start, for then the new chapter became a snow-ball rolling down hill so that on Charter Night, November 1, 1943—a new all-time record as to size of chapter at its start was made by the Indiana Chapter with 81 charter members in attendance, and so became the 32nd Chapter in our long roll of chapters. We had 38 members in Indiana as of September. Since that date 61 new applications have been turned in, or a total of 99 members now in Indiana.

If nothing else had happened during this year, that would have been inspiration enough for all of the work in 1943.

If it can be done in Indianapolis it can also be done in the other cities I have mentioned and several others located strategically throughout the country, like Columbus, Ohio; Newark, N. J.; Salt Lake City, Utah; Syracuse or Central New York; Louisville, Ky.; and a central location in Pennsylvania.

It was proposed this year that all members be assigned to some chapter. I don't think this is practical until we have chapters at least at the various points I have indicated.

From my experience this year I think it is desirable for the president to visit as many of the chapters as it is physically possible for him to do. The chapters look forward to the president's annual visit and the opportunity to make personal contact with the man who is actually the symbol of the Society. Remember that this is the only actual contact many of our members have with the Society, except through

publications. This might be placing a great burden on the president especially as we add more chapters, but it is important to remember that in assuming the great honor of this Office you also accept the responsibility and the need of devoting, except for the summer months, practically your entire time to the work of the presidency.

This year we prepared for each chapter visible index systems for keeping complete records of their members and minute books for keeping standard minutes with duplicate copies for headquarters office. I think we should continue to give more and more assistance to the chapters, particularly in the way of speakers—not limit them to two speakers a year, but make it possible for them to have additional ones with a special message or timely technical information. We could make up a list of topics to be discussed in all the chapters with inter-change of ideas.

It will help build up their morale by making their meetings much more interesting and attract more new members into the society. We get most of our members through the chapters, so why not concentrate on giving them a greater desire to join. The Chapter Relations Committee should be one of our most important Committees, a greater allocation should be provided in its budget so that it can do everything possible to make our chapters stronger and more active units of the Society.

*Fifth—THE GUIDE:* This year THE GUIDE exceeded all expectations, both as to sales and advertising. The Guide Publications Committee will report later with complete factual information.

The Committee is to be congratulated for the splendid work they have done in re-editing THE GUIDE and rewriting the necessary chapters to bring it up to date with current research information.

*Sixth—War Service:* The Society has contributed much to the War effort. I have mentioned the all-out subscription to War Bonds, and we now have 332 members in the Service of the United Nations. Two of our members have paid the supreme sacrifice. I particularly refer to Flying Officer Harold L. Temple, of the Royal Air Force, from London, England and Pilot Officer Phil Foster of the Royal Air Force from the Manitoba Chapter, Canada.

Through our War Service Committee we have contributed much to the National Fuel Rationing Programs, and our individual members throughout the country have rendered yeoman service in helping to make these programs work.

We have been the leaders in the Fuel Conservation Program and practically all our recommendations have been adopted nationally. Our pamphlet on 57 Ways to Save Fuel has been distributed nationally by the Office of Price Administration.

Our individual members have contributed much to the construction period when we were building the Arsenal of Democracy, cantonments, and flying fields all over this hemisphere. The Research Laboratory was early placed at the disposal of the Government and has conducted vital research work for the United States Navy.

If this Administration has achieved any medium of success during the War Year 1943 it has not been because of the work of any one individual, but rather the cooperative efforts of all. I ask the same loyal support for the incoming administration.

I have been blessed with an exceptionally fine Council; the committee chairmen and their committees are all to be commended; the chapters and individual members have all played their part. I wish time would permit me to mention each one by name.

I have had the loyal and faithful cooperation of the New York Staff.

I cannot close without paying special tribute to Secretary A. V. Hutchinson, who seems to never sleep but works day and night for the advancement of the Society and is always at the right-hand of the President, ready to help with all the constant details that require attention from day to day.

It has been a real pleasure and a privilege to serve you for the past year, yes—for the past 25 years. May I continue to do so, and I hope I can be here for the Diamond Jubilee 25 years hence.

Respectfully submitted,

M. F. BLANKIN, *President*

President Blankin referred to the Report of the Council as follows:

#### COUNCIL REPORT

The Council held its organization meeting, January 27, 1943, in Cincinnati, O., and Pres. M. F. Blankin announced the appointment of Council and Special Committees. A. V. Hutchinson was reappointed Secretary. The recommendations of the Committee on Research to have Prof. C. M. Humphreys continue as Acting Director of the Research Laboratory were confirmed. Depositories for Society funds were selected in New York, Brooklyn, and Toronto, and the Finance Committee was authorized to retain a Certified Public Accountant to audit the Society's books.

The budget for 1943 as presented by the Finance Committee provided for an estimated income of \$84,300 and an estimated expenditure of the same amount exclusive of research funds.

Plans for the Semi-Annual Meeting 1943 were discussed and the invitation of the Pittsburgh Chapter was accepted.

At the April 18 Council Meeting in Chicago the Secretary was authorized to prepare a uniform system for keeping Chapter records and minutes and a karderx record for accounts.

Employment of a Technical Secretary was authorized and a special committee was appointed to select a candidate.

Announcement was made of the appointment of Carl H. Flink as Technical Secretary, effective March 1. The activities of the War Service Committee were reviewed by Chairman John Howatt.

The Committee on Research presented five suggestions with reference to the future research program and the Council authorized the creation of the position of Director of Research, and requested the Committee on Research to engage a Director for a three-year term to carry out the program that it had prepared.

The Finance Committee was authorized to determine whether an employees' benefit plan would be desirable. The Committee also submitted the Budget in revised form to conform with the newly installed bookkeeping system.

Two meetings of the Council were held in Pittsburgh, Pa. in June and the Secretary was authorized to negotiate a new lease for headquarters at 51 Madison Ave., New York 10, N. Y., and the recommendations of the Finance Committee were adopted regarding dues rates for members in service. It was voted to invest some of our Canadian Funds in the Fourth Canadian Victory Loan.

A comprehensive plan for handling codes and standards of the Society was submitted by the Standards Committee.

Nominations for members to serve on the Committee on Research were announced and President Blankin appointed the Chapter Development Committee consisting of Messrs. W. A. Russell, Albert Buenger and C. E. Price, to study the chapter problems covered by a resolution of the Northern Ohio Chapter.

Announcement was made of the appointment of Cyril Tasker as Director of Research, effective October 1, 1943.

In November the Council met in Indianapolis, Ind., on the occasion of the organization meeting of the Indiana Chapter, and announced the dues rate for 1944. It authorized the purchase of three *Series F*, U. S. War Savings Bonds and three bonds of the Fifth Canadian Victory Loan.

A change in the fiscal year was adopted and the Finance Committee presented the Budget for the period November 1, 1943 to October 31, 1944 which showed a total estimated income including Research of \$140,878 and total expenditures of \$135,878.

Detailed plans for the 50th Annual Meeting were announced and the Council indicated its desire to have Charter Members and all Past Presidents in attendance. The report on the F. Paul Anderson Award indicated that Lt.-Comdr. F. C. Houghten was the unanimous choice for this award.

The report from the Committee on Research indicated that new quarters for the Laboratory would be required on or before the termination of our present agreement

with the Bureau of Mines and a special committee was appointed to investigate other locations that had been offered.

Life Memberships were granted to ten members who were eligible effective January 1, 1944.

The concluding Council Meeting was held in New York, January 30, 1944 and all pending business was disposed of and final reports of all Council committees were received. During the year routine action was taken on all resignations, cancellations for non-payment of dues, and reinstatement of members.

In review the Council had a year of unusual activity, faced many problems, and made several decisions that will ultimately affect the future of the Society.

Respectfully submitted,

THE COUNCIL

The Report of the Secretary was then submitted as follows:

#### REPORT OF SECRETARY

It is the duty of the Secretary to report on activities of the headquarters office for the past year, and 1943 was crowded with many special projects. Because of the necessity for giving greater service to the members, largely engaged in war activities; cooperation with government agencies confronted with the fuel crisis; many changes in office procedure, required by the installation of new accounting methods; a revised method of handling codes and standards, and the active work of many special committees, made 1943 one of the most unusual years in the Society's history.

It is notable that the Society's records show 334 members in active military service and their names have been placed on the Society's Service Roll which is published periodically. It is unfortunate that there is no way of getting a complete record of the services rendered by the members who are not in uniform, but who are doing such effective work in helping to win the war.

Our records show that a substantial growth of the Society continues and that the present total of 3342 members on the roll is an all-time high. One new chapter was organized during the year and the Indiana group did a phenomenal job of starting November 1 with the largest initial charter membership of any chapter in the society.

It may be interesting to glance backward through the years and take note of the fact that on the occasion of the Society's 25th Anniversary in 1919, the slogan was, "One Thousand Members on the 25th Anniversary Meeting of the Society." This was accomplished with 1058 on the rolls. It is also of interest to read in Transactions that on the third anniversary of the Society in 1897, the President, Prof. R. C. Carpenter, was suggesting a revision of the stringent membership requirements, which he thought would keep the Society exceedingly small. He visualized a national Society at least 500 to 1000 members. The 500 total was not achieved until 1914 and the peak was attained five years later.

As we approach the 50th Anniversary year, it is significant that this also marks the 25th anniversary of the establishment of the Society's Research Laboratory. In 1943, it became apparent that it would be necessary to move in the near future from our present location in Pittsburgh and a special committee was appointed to select a new location for the Laboratory.

In March the Council appointed a new Technical Secretary, Carl H. Flink, and in October the Council created the position of Director of Research and approved the appointment of Cyril Tasker to this position. The activities of both have been carried on at the headquarters office.

In the reports of many committees that will be given at this meeting, it should be possible to visualize the volume of correspondence, publication activity, membership and special services that have been carried on by the headquarters office staff. The growth of the Society to its present high level of activity has created many problems. The present financial position of the Society is sound and analysis of the financial statement prepared by the Society's Certified Public Accountant will

show that dues collections are at a satisfactory level, that reserve funds are adequate and are somewhat above the constitutional requirement.

The Council has held seven meetings during the past year and the Society has held two regular meetings in 1943, at Cincinnati and Pittsburgh. It has been the policy of the Council during the war period to schedule meetings which might be helpful in giving assistance to its members and to government agencies. Programs have been streamlined to meet wartime needs, but it has been noticeable that heavier registrations have been recorded at these meetings, the technical sessions have been better attended and the number of technical committee meetings have greatly increased in volume.

This year, the Council gave special recognition to two of the staff members, Dorothy M. Mildner and Pauline Kurland, who completed 20 years' service with the Society. Although many unusual and difficult problems have been encountered under wartime conditions the staff has endeavored to meet all of the unusual demands and carry on the work effectively. I should like to commend the activities of all my associates of the headquarters office staff who have worked so effectively to meet the exacting demands of 1943 and who greet the 50th year of the Society's service with a desire to make it a memorable one.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary*

President Blankin called upon W. A. Russell, chairman of the Chapter Development Committee, who presented his prepared report as follows:

#### REPORT OF CHAPTER DEVELOPMENT COMMITTEE

The full report of the Chapter Development Committee was printed in an 8 page pamphlet and mailed to the entire membership on January 10, 1944. (See Appendix.)

It will be recalled that this Committee was appointed at the Pittsburgh meeting of the Society in June, 1943, on motion of the Northern Ohio Chapter, and was given the assignment of studying the subject of Chapter-Society relationship.

Your Committee held three meetings—one in Pittsburgh at the time of appointment, one in Indianapolis in September and one in Cincinnati in December. At the first, the general plan of procedure and organization was adopted; at the second, all the details of the subject and the information needed for consideration were thoroughly explored and the decision made to go to the entire membership for an authoritative expression of opinion on the essential problems involved; at the third, the results of our investigations were analyzed, the conclusions drawn up and the report written. Voluminous correspondence between members of the committee and members of the Society supplemented the discussions and studies at the meetings themselves.

All members will recall the letter of explanation and questionnaire sent out to determine the acceptability of either of the two methods of establishing a plan whereby each member of the National Society becomes a member of a local chapter. The specific questions—Are you in favor of chapter subsidy with increased national dues?—and—Are you in favor of chapter subsidy without increased national dues but with curtailed services to members?—were asked.

Every member was sent the questionnaire. The very high return of 50 per cent was the result. Based upon the returns—which, according to all research experience, are more than ample to accept as a reflection of the wishes and opinions of all members—68 per cent of the membership does not want an increase in national dues for the purpose of chapter subsidy; 30 per cent does; 2 per cent has no opinion; and 83 per cent of the membership is opposed to curtailing present Society services in order to subsidize chapters; 14 per cent is for it; 3 per cent has no opinion.

Before asking your acceptance of this report, your Committee would like to refer to Article B-IV—Admission Fees and Dues in the Constitution, By-Laws and Rules of the Society. The admission fee and annual dues are there prescribed as "determined by the Council until 1945 and thereafter . . ." the dues shall be \$25.00, etc. At that time, of course, the feasibility of such a plan as we have been asked to study would assume a different aspect, but the Committee calls to the Council's attention,

as perhaps bearing on the general subject of dues, this decisive note against increased dues. While the opposition to them is in connection with a plan for chapter subsidy, it may or may not be construed as opposition to increased dues for any reason.

Your Committee on Chapter Development now requests confirmation of its conclusion that any plan for Chapter subsidy by the National Society is not feasible at this time and acceptance of its report; and it respectfully asks that it be discharged as a committee of the Society. I so move.

Respectfully submitted,

W. A. RUSSELL, *Chairman*  
ALBERT BUENGER,  
CHAS. E. PRICE.

## Appendix

### REPORT OF CHAPTER DEVELOPMENT COMMITTEE

JANUARY 1944

The Chapter Development Committee—W. A. Russell, chairman, Albert Buenger and Chas. E. Price—was, on motion presented by the Northern Ohio Chapter, appointed at the Pittsburgh meeting of the Society, June, 1943, to study the subject of Chapter-Society relationship and specifically to investigate:

1. A plan whereby each member of the National Society becomes a member of a local chapter;
2. A plan for the geographical boundaries of each chapter, thus determining its membership on a convenient geographical basis;
3. A schedule of national dues and chapter dues, taking into consideration the possible financial support of the chapters by the National Society;
4. Such other matters that may be considered pertinent to this subject.

It was pointed out in the motion of the Northern Ohio Chapter that, under present conditions, National Society members may not belong to a local chapter unless they separately apply for admission and pay local chapter dues; local chapters may not admit to membership anyone not a member of the National Society; the National Society seeks the help of chapters in securing new members, serving as hosts at meetings, carrying on Society activities; and the continued growth and strength of the National Society are directly related to the growth and strength of the several chapters.

This matter of Chapter-Society relationship has come before the Society on several occasions in past years. Other committees have studied it from various angles. Throughout all such past studies recognition of the value of chapters has been clearly emphasized, and a desire to do all possible to establish new chapters and make all chapters strong has been similarly apparent. Out of such former committees' recommendations have come the National Speakers' Bureau, the formation of a Chapter Delegates' Committee and the payment each year of chapter delegates' railroad fares to and from the annual meeting of the Society. These services have been established and are financed by the National Society for the benefit of chapters.

Thus, the present committee has concentrated its investigation on the possibilities of making every Society member a chapter member and of financial support for the chapters by the National Society. If these were feasible, then and only then would it be necessary to go into the other assignments such as dues schedules, geographic boundaries of chapters and other matters pertinent to the subject.

While trying to arrive at a basis for judgment of these possibilities, it assembled data on what other engineering societies did in these regards. Through the cooperation of L. T. Avery, Cleveland, it received information that a number of these other leading societies do automatically make their national members local chapter or section members and do rebate dues to chapters or provide other financial support or both. In fact, we uncovered none which does not have such a policy. The A.S.H.V.E. is an exception among national engineering societies in not having this kind of a set-up.

In the committee's discussions, however, it quickly became apparent that, no matter what other societies did, it was necessary to determine exactly what such a plan for the A.S.H.V.E. would entail.

The first fact it wanted was whether the present \$18.00 national dues permitted a rebate of dues to chapters for every member, if this plan were adopted. A study of chapter dues' schedules revealed that local dues now range from \$2.00 to \$7.00. Three-fourths of the chapters have dues between \$3.00 and \$5.00. Thus, it was apparent that a plan to rebate chapter dues would cost money. It was apparent, too, that the amount of rebate, if the plan were to be put in operation, would be a problem, for high-rate chapters could not operate on a rebate which would satisfy low-rate chapters.

The question of the ability of the National Society to finance such a program out of present dues was, therefore, presented to the proper officials and your committee was advised that Society income solely from dues was insufficient. Figures supporting this statement were given to your committee. In the years 1939-1942 your Society has failed to meet operating expenses out of dues. There has been an average yearly loss of \$820.00.

Other Society income, such as surplus from THE GUIDE, initiation fees and interest on investments, is earmarked, by the by-laws and rules of the Society, for research, endowment and reserve funds. To use any part of it for some form of chapter subsidy would mean that some of the Society's present research plans and services to members would have to be discontinued or curtailed.

Thus, it was evident that a plan whereby each member of the National Society becomes a member of a local chapter, which would involve financial support from the National Society, must be accompanied by an increase in the national dues of every member or by a curtailment of present Society services and activities.

Your committee did not feel it could, on its own, recommend or condemn a plan which called for action affecting both the cost and the benefits of Society membership. Through a questionnaire it sent to chapter delegates, in August, 1943, it determined that from 30 to 40 per cent of the national membership is not in local chapters at the present time. Such non-chapter members should obviously be questioned as to their willingness to pay more dues or accept less service from national membership for the privilege of chapter membership. Also, chapter members should be questioned as to their willingness to change their present methods of chapter operation.

Thus, in October, 1943, the committee addressed a letter and post-card questionnaire to every member of the National Society. The letter was aimed to explain briefly the problems involved in this whole matter so that each member would have the proper background for his judgment. The questionnaire was as follows:

### QUESTIONNAIRE

#### Chapter Development Committee, A.S.H.V.E.

Please fill out and mail at once.

1. Did you read the accompanying letter? Yes..... No.....
2. Are you now a chapter member? Yes..... No.....
3. Are you in favor of chapter subsidy with increased National Dues? Yes..... No.....
4. Are you in favor of chapter subsidy without increased National dues but with curtailed services to members? Yes..... No.....
5. If you are not now a chapter member is it possible for you to belong? Yes..... No.....
6. If you are not now a chapter member and are near a chapter headquarters why are you not a member? .....

Comments .....

.....

.....

Signature .....

(Optional)

It was impossible, under the circumstances outlined, to ask—are you in favor of a plan whereby each member of the National Society becomes a member of a local chapter—without determining, first, if a method by which it could be put into operation was acceptable or not acceptable to members. It, therefore, asked the two essential questions—(a) are you in favor of chapter subsidy with increased national dues; (b) are you in favor of chapter subsidy without increased national dues but with curtailed services to members—either one of which had to be answered in the affirmative by a majority of the members if this plan were to be put into effect. It provided room for comments. It also asked the questions about chapter membership with the hope that some information of help to chapter operations would result.

To avoid any possibility of incomplete or biased returns, we addressed it to *every* member of the Society. Research experts are agreed that a 10 per cent sampling, if the proper samples are taken, will accurately reflect the opinion of the whole. Gallup and others say that even less is required. We have seen a sample of 27 people set up as an accurate, reflection of the views of 15,000,000. But we wanted every member to have a voice in the decision we were asked to bring before this meeting. We took a 100 per cent sample, 3,126 letters and questionnaires were sent out.

Good returns for a mail questionnaire on any subject are 10, 15, 20 and 25 per cent. They are accepted as satisfactory in all market research. As evidence, when half the returns of this questionnaire were in, we tabulated them and the results were identical with the final returns from the whole. But we don't have to accept a small return. We heard from 1,533—50 per cent of the membership. We do not believe that any A.S.H.V.E. activity, including election of officers, has ever had so many members participating. So, there should be no question about interpreting the results from this questionnaire as reflecting the wishes and opinions of the membership.

The replies were tabulated by chapter members, non-chapter members and by states. Accompanying this report is a complete summary of replies. 67 per cent of the returns were from chapter members; 33 per cent were from non-chapter members. Every state in which there is a national member was represented, including Canada.

Out of 1,533 replies, 1,052 voted "no" to chapter subsidy with increased national dues; 461 voted "yes"; 20 did not answer this particular question. In other words, 68 per cent of the membership does not want an increase in national dues for the purpose of chapter subsidy; 30 per cent does; 2 per cent has no opinion.

The vote of chapter members was—1,021 replies: 699 "no"; 313 "yes"; 9 did not answer. 68 per cent of the present chapter members, therefore, is opposed to increasing national dues for the purpose of chapter subsidy; 31 per cent is for it; 1 per cent has no opinion.

The vote of non-chapter members was—512 replies: 353 "no"; 148 "yes"; 11 did not answer. 69 per cent of present non-chapter members is opposed to chapter subsidy with increased dues; 29 per cent is for it; 2 per cent has no opinion.

Thus, returns from chapter and non-chapter members are remarkably parallel, showing that no one division of the membership feels any different about this proposal than the other.

The states of Arkansas, Delaware, Florida, Georgia, Nevada and Wyoming were the only ones in which there were majorities in favor of increasing national dues for chapter subsidy. The total votes from these states were 40 out of the 1,533 received. 25 voted "yes"; 14 voted "no." There were 96 members of the Society in these states.

In Alabama, South Dakota, Texas, Utah and West Virginia, the votes were equally divided. Two members each from Alabama, South Dakota, Utah and West Virginia voted. In each case, one voted "yes"; one voted "no." 53 from Texas voted. 26 voted "yes," 27 voted "no."

In all other states, the majority of members voted "no." In Ohio, from one of whose chapters this motion came, 113 out of 185 members voted. 69, or 61 per cent, voted "no," 43, or 38 per cent, voted "yes"; one had no opinion.

The votes for a chapter subsidy without increased national dues but with curtailed services to members were predominantly "no." Out of 1,533 votes, 1,277 were "no," 211 were "yes," 45 had no opinion. Thus, 83 per cent of the membership is opposed

to curtailing services in order to subsidize chapters; 14 per cent is for it; 3 per cent has no opinion.

Votes of chapter members were 855, 84 per cent, "no"; 143, 15 per cent, "yes"; 23, 2 per cent, no opinion.

Votes of non-chapter members were 422, 82 + per cent, "no"; 68, 13 per cent, "yes"; 22, 4 per cent, no opinion.

Many interesting comments were received. Those who were against increasing dues and curtailing services to permit a plan whereby every national member becomes a chapter member generally felt that each chapter should pay its own expenses. That local membership should be solicited locally and that chapters should earn and deserve the interest of members, and that there be nothing compulsory about chapter membership.

Some felt this was a post-war problem and that when there were more chapters available to all members there might be merit in this plan. Many non-chapter members, of course, pointed out their distance from chapters and their inability to attend meetings for that reason.

Those who were for the plan, either by increasing dues or curtailing services, felt that payment of national dues should entitle the member to all rights in the local chapter. Many specified that an increase of \$1.00 or \$2.00 should be the limit. Many referred to other engineering Societies in this regard and felt that the A.S.H.V.E. should do the same. There were some who felt that such a plan could be put into effect without increasing dues or curtailing services.

However, the committee feels that the membership has spoken with full knowledge of the problem and its benefits or short-comings. Better than two to one are opposed to increasing dues, six to one are opposed to curtailing services.

Thus, a plan whereby each member of the National Society becomes a member of a local chapter is not feasible, inasmuch as the only two methods by which such a plan could be put in operation are not acceptable to the membership.

This being true, your committee, has not endeavored to set up a plan for the geographical boundaries of each chapter. It has, for the same reason, no plans for a schedule of national dues and chapter dues.

As for other matters that may be considered pertinent to this subject, the committee has analyzed these returns to determine how many members voted "no" to both questions. This number was 826, 54 per cent. It might be interpreted, therefore, that 46 per cent of the membership is in favor of some plan for making each national member a chapter member. This indicates more support for the general idea than the statistics on each method of financing. In other words, we have a situation where nearly half the membership wants something done about Chapter-Society relationship, but more than two-thirds don't want to increase dues to obtain it and more than four-fifths don't want to discontinue or curtail any present Society services to obtain it.

It is evident, too, from the replies to the other questions asked in this survey that chapters can improve their membership and standing through more active membership drives and more interesting meetings. Out of the 512 non-chapter members, 241 state it is possible for them to belong to a chapter. The reasons why they are not now members are (1) they have not been asked; (2) they do not feel the meetings are sufficiently informative and interesting, to justify belonging; (3) they have not felt they had the time. Thus, on the average, every chapter could, through the proper work and interest, obtain perhaps 40 to 50 per cent of the non-chapter members in its area as members.

It is apparent, too, from these replies that there will always be Society members who feel they can not belong to chapters. The majority, of course, are those who are remote from chapters, but there are many who live in or near chapter cities who travel constantly or who have other engagements on chapter meeting nights.

## TYPICAL COMMENTS

*Question 3—Are you in favor of chapter subsidy with increased National dues? Yes..... No.....*

**Chapter members, voting Yes:—**

1. "Raise Society dues to \$20.00 per year and pay \$2.00 per year to chapter assigned to member. Allow member to choose chapter subject to approval."
2. "This is a real forward step to get national Society benefits to present 'local members.'"
3. "I feel that compulsory 'National' dues for benefit of chapter treasuries would probably make for stronger chapters (more members) and possibly reduce cost of dues by the same token."

**Non-chapter members, voting Yes:—**

1. "Should be a rebate to members in locations remote from chapter city, or where it is impossible to attend meetings for other compelling reasons."
2. "Am in favor of #3 after the war; however owing to gas rationing do not favor any action now."

**Chapter members, voting No:—**

1. "It is possible that a member may only desire National and not chapter benefits. He should not be forced to contribute directly or indirectly to chapter activities, particularly if he is so remote he cannot attend."
2. "Any increase in dues at this time is likely to have serious effect on present membership. If a member is not sufficiently interested in activity of chapter he is better off not being forced into it."
3. "If a chapter makes its meetings sufficiently attractive it needs no subsidy. (Subsidy is an obnoxious word, why not say 'Dole?')"

**Non-chapter members, voting No:—**

1. "Have no desire to become a member and if it is made compulsory I will drop out of the Society."
2. "I think that local chapter affiliation should be entirely a personal matter."

## TYPICAL COMMENTS

*Question 4—Are you in favor of chapter subsidy without increased National dues but with curtailed services to members?*

*Yes..... No.....*

**Chapter members, voting Yes:—**

1. "Prefer to see dues remain at present level and some means devised to make plan No. 4 above workable."
2. "Even with immediate curtailed services to members I think final results will benefit National Society. Chapters can turn subsidy back if they do not need it."
3. "Keep dues as they are. Cut out sending A.S.H.V.E. TRANSACTIONS and GUIDE as standard. Let members buy these if they want them and pay extra."

**Non-chapter members, voting Yes:—**

1. "Depending on which services are curtailed."
2. "The National dues should cover chapter dues, I feel, and consequently do not belong to a local chapter."

**Chapter members, voting No:—**

1. "Chapter subsidy may not require any curtailment of services."
2. "I do not believe the National chapter should ever decrease its service to members. I believe in individual local chapter maintenance if the National chapter is to be sacrificed."
3. "The foundation of our Society is research without which we have no foundation."

**Non-chapter members, voting No:—**

1. "Such extra dues would be unfair to those not close enough to a chapter to attend the meetings. Society activities should not be curtailed."
2. "Am not in position or find time to attend meetings. Cannot see where anything is gained by compulsory chapter membership. Do not like the word compulsory."

## TYPICAL COMMENTS

*Question 6—If you are not now a chapter member and are near a chapter headquarters why are you not a member?*

1. "Have just contacted local chapter and have attended one meeting. Think I will join."
2. "Procrastination. The local chapters should be more alert and active in soliciting membership."
3. "I live 40 miles from the nearest chapter and the transportation to and from are bad and can not get gas to drive at present."
4. "Because of traveling."
5. "National dues are enough to put out in this direction."
6. "Desire to aid National organization work but do not intend to become active in local chapter or affiliated with it."
7. "Resigned. Did not approve of the chapter politics or policies. Think a mistake to force members to belong to local chapters. Suggest that closer National supervision of chapters would make them more attractive to local engineers."
8. "Chapter services do not seem to be sufficient to warrant membership."
9. "I find so little of interest at the chapter meetings. If the chapter would arrange real Technical meeting I would gladly belong. Chapter meetings are too much 'sales' talk."
10. "Frequent moving makes chapter membership impractical. (Moving due to War Conditions.) Will rejoin chapter at point of settling after completion of war duties."
11. "Too busy in past years. Probably will affiliate next year."
12. "Nearest chapter 180 miles.—Am in Army now but hope to join chapter in Washington, D.C., after war."

## TABULATION OF REPLIES

*Question 3—Are you in favor of chapter subsidy with increased National dues?*      *Question 4—Are you in favor of chapter subsidy without increased dues but with curtailed services to members?*

U. S. AND CANADA	TOTAL MEMBERS	TOTAL REPLIES	TOTAL NO	TOTAL YES	TOTAL NO ANSWER	TOTAL NO	TOTAL YES	TOTAL NO ANSWER
Chapter								
Members		1021	699	313	9	855	143	23
Non-chapter								
Members		512	353	148	11	422	68	22
Total	3126	1533	1052	461	20	1277	211	45
STATE	TOTAL MEMBERS	TOTAL REPLIES	TOTAL NO	TOTAL YES	TOTAL NO ANSWER	TOTAL NO	TOTAL YES	TOTAL NO ANSWER
Alabama								
Chapter		0	0	0	0	0	0	0
Non-chapter		2	1	1	0	2	0	0
Total	8	2	1	1	0	2	0	0
Arkansas								
Chapter		0	0	0	0	0	0	0
Non-chapter		3	1	2	0	3	0	0
Total	6	3	1	2	0	3	0	0
California								
Chapter		54	26	27	1	44	9	1
Non-chapter		25	20	5	0	20	4	1
Total	129	79	46	32	1	64	13	2
Colorado								
Chapter		1	1	0	0	0	1	0
Non-chapter		3	3	0	0	3	0	0
Total	8	4	4	0	0	3	1	0
Connecticut								
Chapter		14	12	2	0	11	3	0
Non-chapter		8	5	3	0	7	1	0
Total	40	22	17	5	0	18	4	0
Delaware								
Chapter		4	1	2	1	4	0	0
Non-chapter		2	1	1	0	2	0	0
Total	10	6	2	3	1	6	0	0
Dist. of Col.								
Chapter		46	31	14	1	41	5	0
Non-chapter		19	14	4	1	16	3	0
Total	88	65	45	18	2	57	8	0
Florida								
Chapter		1	0	1	0	1	0	0
Non-chapter		8	3	5	0	7	0	1
Total	16	9	3	6	0	8	0	1

STATE	TOTAL MEMBERS	TOTAL REPLIES	TOTAL NO	TOTAL YES	TOTAL NO ANSWER	TOTAL NO	TOTAL YES	TOTAL NO ANSWER
Georgia								
Chapter		16	7	9	0	12	4	0
Non-chapter		4	1	3	0	4	0	0
Total	57	20	8	12	0	16	4	0
Illinois								
Chapter		80	55	24	1	65	14	1
Non-chapter		52	37	14	1	40	9	3
Total	249	132	92	38	2	105	23	4
Indiana								
Chapter		20	16	4	0	17	2	1
Non-chapter		7	6	1	0	7	0	0
Total	38	27	22	5	0	24	2	1
Iowa								
Chapter		13	12	1	0	11	2	0
Non-chapter		4	4	0	0	2	0	2
Total	39	17	16	1	0	13	2	2
Kansas								
Chapter		4	2	2	0	2	2	0
Non-chapter		5	5	0	0	5	0	0
Total	15	9	7	2	0	7	2	0
Kentucky								
Chapter		2	2	0	0	2	0	0
Non-chapter		4	3	1	0	4	0	0
Total	12	6	5	1	0	6	0	0
Louisiana								
Chapter		14	10	4	0	11	3	0
Non-chapter		2	2	0	0	2	0	0
Total	36	16	12	4	0	13	3	0
Maine								
Chapter		0	0	0	0	0	0	0
Non-chapter		2	2	0	0	2	0	0
Total	4	2	2	0	0	2	0	0
Maryland								
Chapter		4	4	0	0	4	0	0
Non-chapter		12	10	2	0	10	2	0
Total	58	16	14	2	0	14	2	0
Massachusetts								
Chapter		32	21	11	0	28	3	1
Non-chapter		18	10	7	1	14	3	1
Total	100	50	31	18	1	42	6	2
Michigan								
Chapter		75	61	14	0	66	8	1
Non-chapter		27	18	8	1	23	2	2
Total	182	102	79	22	1	89	10	3
Minnesota								
Chapter		43	38	5	0	37	4	2
Non-chapter		8	5	3	0	7	1	0
Total	93	51	43	8	0	44	5	2
Mississippi								
Chapter		0	0	0	0	0	0	0
Non-chapter		3	2	1	0	2	1	0
Total	4	3	2	1	0	2	1	0
Missouri								
Chapter		61	44	17	0	48	10	3
Non-chapter		13	10	2	1	10	2	1
Total	110	74	54	19	1	58	12	4
Montana								
Chapter		0	0	0	0	0	0	0
Non-chapter		1	1	0	0	0	1	0
Total	3	1	1	0	0	0	1	0
Nebraska								
Chapter		11	7	4	0	11	0	0
Non-chapter		2	2	0	0	1	1	0
Total	21	13	9	4	0	12	1	0
Nevada								
Chapter		0	0	0	0	0	0	0
Non-chapter		1	0	1	0	1	0	0
Total	5	1	0	1	0	1	0	0
New Jersey								
Chapter		20	13	7	0	17	3	0
Non-chapter		24	17	7	0	20	2	2
Total	112	44	30	14	0	37	5	2
New York								
Chapter		113	74	36	3	96	15	2
Non-chapter		72	46	23	3	54	16	2
Total	380	185	120	59	6	150	31	4
North Carolina								
Chapter		20	15	5	0	18	2	0
Non-chapter		1	1	0	0	0	1	0
Total	43	21	16	5	0	18	3	0

STATE	TOTAL MEMBERS	TOTAL REPLIES	TOTAL NO	TOTAL YES	TOTAL NO ANSWER	TOTAL NO	TOTAL YES	TOTAL NO ANSWER
Ohio								
Chapter		80	47	33	0	61	17	2
Non-chapter		33	22	10	1	25	7	1
Total	185	113	69	43	1	86	24	3
Oklahoma								
Chapter		5	4	1	0	5	0	0
Non-chapter		4	3	1	0	4	0	0
Total	15	9	7	2	0	9	0	0
Oregon								
Chapter		19	15	4	0	13	5	1
Non-chapter		1	0	1	0	1	0	0
Total	43	20	15	5	0	14	5	1
Pennsylvania								
Chapter		90	64	26	0	76	12	2
Non-chapter		43	31	11	1	37	5	1
Total	248	133	95	37	1	113	17	3
Rhode Island								
Chapter		1	1	0	0	1	0	0
Non-chapter		3	3	0	0	3	0	0
Total	7	4	4	0	0	4	0	0
South Carolina								
Chapter		0	0	0	0	0	0	0
Non-chapter		3	2	1	0	3	0	0
Total	12	3	2	1	0	3	0	0
South Dakota								
Chapter		0	0	0	0	0	0	0
Non-chapter		2	1	1	0	2	0	0
Total	2	2	1	1	0	2	0	0
Tennessee								
Chapter		1	0	1	0	1	0	0
Non-chapter		10	7	3	0	9	1	0
Total	17	11	7	4	0	10	1	0
Texas								
Chapter		38	18	20	0	31	6	1
Non-chapter		15	9	6	0	12	1	2
Total	86	53	27	26	0	43	7	3
Utah								
Chapter		1	0	0	1	0	0	1
Non-chapter		2	1	1	0	2	0	0
Total	8	3	1	1	1	2	0	1
Virginia								
Chapter		6	6	0	0	5	1	0
Non-chapter		12	11	1	0	11	1	0
Total	55	18	17	1	0	16	2	0
Washington								
Chapter		16	9	7	0	13	3	0
Non-chapter		6	5	1	0	6	0	0
Total	49	22	14	8	0	19	3	0
West Virginia								
Chapter		0	0	0	0	0	0	0
Non-chapter		4	2	2	0	3	1	0
Total	8	4	2	2	0	3	1	0
Wisconsin								
Chapter		36	27	8	1	29	5	2
Non-chapter		15	14	0	1	13	0	2
Total	88	51	41	8	2	42	5	4
Wyoming								
Chapter		0	0	0	0	0	0	0
Non-chapter		1	0	1	0	1	0	0
Total	2	1	0	1	0	1	0	0
United States								
Chapter		941	643	289	9	781	139	21
Non-chapter		486	341	134	11	400	64	22
Total	2691	1427	984	423	20	1181	203	43
Canada								
Chapter		75	54	21	0	69	4	2
Non-chapter		19	8	11	0	17	2	0
Total	218	94	62	32	0	86	6	2
APQ, Navy, etc.								
Chapter		3	2	1	0	3	0	0
Non-chapter		6	4	2	0	4	2	0
Total		9	6	3	0	7	2	0
Unidentified								
Chapter		2	0	2	0	2	0	0
Non-chapter		1	0	1	0	1	0	0
Total		3	0	3	0	3	0	0

F. W. LEGLER, Minneapolis, Minn.: Due to the fine work of the Chapter Development Committee, and the complete report that was sent out to all members, I cannot understand why any discussion is necessary, because it has been so thoroughly covered by 50 per cent of the membership, which certainly indicates the general will of the membership. No discussion here would change the viewpoint of the membership, would it? I think it should be adopted and the committee discharged with thanks.

M. F. BLANKIN, Philadelphia, Pa.: Mr. Legler, the request for the appointment of this committee originally came from a resolution presented through L. T. Avery, Cleveland, Ohio, from the Northern Ohio Chapter, and I think they wish to present

some facts in connection with it and have asked permission to discuss it, so I think it is only fair to extend that privilege.

MR. AVERY: Your committee has approached the subject in a truly democratic spirit. They tried to refer this back to the people. As sponsor of the resolution that inspired the appointment of the committee, it is fair to say that I was consulted as to the personnel of the committee and heartily approved the selection of Messrs. Russell, Price and Buenger, who have really worked on this assignment. As a matter of fact, this Society gets a lot of work done by able and busy men.

A problem recognized is a problem half solved. If you recall, at Pittsburgh when I presented the resolution, I also presented some figures on membership of our Society and that of other similar societies. The report given by Mr. Blankin, showing that a membership which had been reduced in four years from 3,300 to 3,000 without anybody paying much attention and which has jumped back to the 3,300 again, shows it was worth raising the question. Mr. Blankin decided that this business of membership had nothing to do with Chapter support. He went out to prove it, and he has done an excellent job.

In this report the membership is opposed to increasing dues and is opposed to curtailing services. Its seems to me that result could have been anticipated from the wording of the post card questionnaire and the letter which accompanied it. The very word *subsidy* is a bad word. *Compulsory* is a bad word, and yet almost one-half the voters wanted something done.

May I question the statement in the report to the effect that the Society could not afford to pay any part of Chapter expense without raising dues or curtailing present Chapter services? You men who voted *no* on the questionnaire were not given the third—the \$64 question: "Are you in favor of Chapter subsidy without increased national dues?"

The implications in the report are grossly unfair to our very able management. While our Society does not always make money each year, it has shown a surplus during the past ten years. Being a technical Society organized not for profit, it would be sinful to make a profit so the By-Laws provide that the surplus each year be allocated to Research or Reserve Funds, or both.

I do not wish to criticise the management that has so ably conducted the affairs of this Society. A year ago in Cincinnati when I served on the Constitution and By-Laws Committee, I said there could not be so much wrong with our method of nominating officers, as we had been blessed with very fine ones. You should know, however, that the reason they claim there is no money for chapter development is that the surplus funds, are allocated elsewhere.

Specifically, the Constitution requires 40 per cent of the dues and all the surplus from THE GUIDE to be allocated to the Research Fund. Each year additional funds from the Society's general income are allocated to research over and above these constitutional requirements. Likewise, a Reserve Fund of \$15 per member is required by the Constitution and By-Laws, which Fund has been given me as of September 15, as \$55,948, or a little better than \$16 per member at the present time. Incidentally, no further funds need be added to that fund, because it is now over its constitutional requirements. No one would complain that the fund is too large, but that is one place where our surplus money has gone.

Your Society has other Reserve and Endowment Funds, so that altogether you have accumulated \$128,941.75 in various funds, which funds now become an investment problem.

I bring up these facts, as the \$64 question could have been asked without jeopardy to our Society even if you had voted *Yes*.

Let me ask you the question in another way: do you think any amount of money, say \$1,500 or \$3,000 annually, from the surplus funds of the Society as at present operated, will do more good if used for chapter support than if added to Reserve Funds or Research Funds?

Have you given thought to the possibility that closer cooperation and support of the Chapters would increase the membership of the Society—hence its income? That was one of the things I hoped to accomplish by this survey—namely, to show an increase in membership without *costing* anything.

For example, I suggested one dollar per member, or possibly \$100 per year per chapter, thus making a budget figure of less than \$3,000 per year.

It is easier to sell one membership than two, so I would expect a very real increase in membership, if, when you joined the Society, you automatically joined the local Chapter in that locality. Is the chapter a social club, or is it part of the A.S.H.V.E.?

Forty per cent of the \$18 dues is directly allocated to research, so the Society General Fund gets the balance, \$10.80 per member. The extra cost of servicing this member with THE GUIDE, *Heating, Piping and Air Conditioning*, and TRANSACTIONS, is about \$4, leaving \$6.80 for general administrative overhead. Thus 500 new members would pay in \$3,400, sufficient for the suggested Chapter Development Fund, and at the same time \$3,600 is added to research.

Do you think 500, 1,000, or 2,000 additional members would be secured through the proper support of the chapters?

President Blankin has done a magnificent job this year in bringing in new members. Are we giving him the best selling program?

If any society is to depend on its chapters for the selection of national officers, and for securing new members, it should have a very real interest in and control of those chapters, which would not permit a chapter to dissolve without our knowledge or consent, so that it ceased to hold meetings.

Any of you could predict the general trend of this particular poll as soon as you read the post card. But you cannot laugh off the fact that almost one-half of the voters wanted something done, even though it cost more money or curtailed some present services. How many more would have favored Chapter support if they thought it would cost nothing?

I think we should accept the committee's report and discharge the committee with our sincere thanks.

PRESIDENT BLANKIN: Thank you very much, Mr. Avery, I am sure this whole thing has done a tremendous amount of good. The Northern Ohio Chapter is also to be thanked for bringing this up, and I know something will develop from that. I think I touched upon it a little bit in my report.

MR. LEGLER: I think the membership might be interested in a discussion which was held at a meeting of the Minnesota Chapter on this very subject, one thing was brought out which probably had a bearing on the outcome of this survey, and that was the implied control which the Society would have in the disbursement of Chapter money. In other words, the Minnesota Chapter was afraid that if this sort of thing was adopted here, we were going to be told what to do with that money and how to spend it. I just wanted to bring this out because to us that was a very important factor.

On motion of W. A. Russell, seconded by C. E. Price, it was voted that the Chapter Development Committee report be accepted and the committee discharged with thanks for their painstaking and effective work.

A detailed report of the Guide Publication Committee was given by P. D. Close, Chairman, who stated that while wartime paper conservation measures had prevented any expansion of the text of the 22nd edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE, it had been possible to add desirable new material by condensing present Guide text without sacrificing essential data.

The Report of the Guide Publication Committee was received with a rising vote of thanks.

#### REPORT OF TELLERS

R. A. Wasson, New York, N. Y., chairman of the Board of Tellers, presented the results of the ballot for officers and members of the Committee on Research as reported on the following page.

## BALLOTS FOR OFFICERS

Total Ballots received.....	1020
Total Legal Ballots.....	1002
President—S. H. Downs.....	1000
First Vice-President—C. E. A. Winslow.....	998
Second Vice-President—Alfred J. Ofner.....	999
Treasurer—L. P. Saunders.....	1002
<i>Members of Council (three-year term):</i>	
C. M. Ashley.....	999
L. T. Avery.....	998
L. E. Seeley.....	998
G. D. Winans.....	1001
(Scattering votes for other candidates.)	

## BALLOTS FOR COMMITTEE ON RESEARCH

Total Ballots received.....	1020
Total Legal Ballots.....	1002
<i>Three-Year Term:</i>	
C. M. Ashley.....	999
F. E. Giesecke.....	1002
F. C. McIntosh.....	1001
G. L. Tuve.....	1002
T. H. Urdahl.....	1001
<i>Two-Year Term:</i>	
John James.....	999
(Scattering votes for other candidates.)	

President Blankin stated that 1020 ballots had been cast for Officers, Council members, and for members of Committee on Research, which indicated a very real interest on the part of the members of the Society.

President Blankin expressed the Society's appreciation of the kind congratulatory messages and expressions of best wishes which had been presented.

The first session adjourned at 12:15 p.m.

## SECOND SESSION—MONDAY, JANUARY 31, 2:30 P.M.

The second session was called to order by President Blankin at 2:30 p.m. in the Georgian Room.

The report of the Committee on Research for 1943 was presented by Chairman C. M. Ashley, Syracuse, N. Y.

## Twenty-Five Years of Research

*Introduction*

This year the Society completed 25 years of research and, whilst today the research activities of the Society are the accepted order of things, 25 years ago they were a radical innovation.

The Research Laboratory was officially instituted on August 1, 1919, under Dean John R. Allen as director. It was located in the Bureau of Mines Building in Pittsburgh, Pa., with a staff of three, one of whom was on the payroll of the Bureau of Mines. The constant temperature room, located in the basement of the Bureau of Mines building, in which so many of the Society's studies on the effect of environmental conditions on man have been conducted, was completed during 1920.

The first cooperative contract was also established in 1919 at the University of Minnesota. For those who know the long distinguished and continuing research record of Prof. F. B. Rowley, it is not surprising that the work was carried on under his direction. The study was concerned with radiant heat losses from direct radiators.

During the past 25 years research has been carried on at the Laboratory on a wide range of subjects; paralleling this has been the work of the Committee on Research, the many technical advisory committees, and work at a large number of cooperating institutions. These all complete a picture of research activity which has contributed much to the growth and present stature of the Society and of the Industry.

### Report of Committee on Research

In common with most similar organizations, the activities carried on by the Committee on Research have, of necessity, been curtailed due to the diversion of attention and personnel caused by the war effort. As a result a number of the Technical Advisory Committees were relatively inactive and the cooperative research at universities was sharply curtailed. The Society is under a debt of gratitude to C. M. Humphreys for adding to the burden of his normal teaching duties at Carnegie Institute of Technology, the part-time direction of the activities at the Research Laboratory.

The chief accomplishments of the Committee on Research during the past year have been in the field of organization and plans for the future. Early in the year plans were formulated looking not only toward bringing the research activity back to its former rate, but also increasing the scope and magnitude of the research operations. These plans visualized the following specific steps:

1. Secure a Director of Research for the Society who would give his full time and attention to the research activity of the Society, and who would be in administrative charge of all of the research activities, including both those at the Laboratory of the Society and those carried on in cooperating institutions.
2. Propose amendments to the Regulations under which the Committee on Research operates so as to permit more efficient administration.
3. Secure a new home for the Research Laboratory of the Society which would provide adequate facilities and room for future expansion, in keeping with the great possibilities of the Society's research program.
4. Augment the laboratory staff commensurate with the enlarged program.
5. Develop a long-range program of research, oriented primarily to the needs of the Industry and engineering profession served.
6. Encourage the more extensive participation of Industry in the research program through both financial contributions and program recommendations.
7. Enlarge the cooperative research activity carried on by the universities as their facilities permit.
8. Encourage the coordination of all the technical activities of the Society.

The first three steps of this program have already been taken. On October 1, Cyril Tasker became Director of Research. The effect of his vigorous leadership is already evidenced in the quickened tempo of the interest and activities of the Committee on Research. Announcement has been made of the plans for the new research laboratory which, it is believed, will serve as the cornerstone of a greater research structure. At this meeting you are being asked to approve proposed amendments to the Regulations Governing the Committee on Research.

Much still remains to be done to consummate the other steps of the plan, but here also the ground work has been laid. The Committee on Research believes that we are well on the way toward the realization of the latent potentialities existing in the program as visualized.

During the year, two research programs being conducted on behalf of the U. S. Navy Department were successfully completed. One of these having to do with physiological reactions was carried on at the Society laboratory, while the second concerning coil performance was conducted at Case School of Applied Science. Members of the Society can take some satisfaction not only in the creditable work done, but also in the fact that the facilities and basic staff of the Society laboratory were contributed gratis for the program. While the research work both for the laboratory and of the cooperating institutions was considerably restricted, there were significant contributions to basement heat loss, air distribution, filter performance, cooling tower performance, psychrometric properties, panel heating, heat gain through sunlit walls, etc. Much significant work was done during the year which has not yet been published. Eleven cooperative research projects were active; three meetings of the Committee on Research were held; five meetings of the Research Executive Committee; twenty-eight meetings of the Research Technical Advisory Committees.

The financial highlights of the research operations for the period ending October 31, 1943, are as follows:

Total Income.....	\$46,840
Total Expenses.....	37,850
Excess of Income over Expenses.....	8,990

## The Income was made up of:

Dues from members and associate members.....	\$ 16,900
Special Council Appropriation.....	5,000
General Contributions.....	5,790
Earmarked Funds.....	19,150
(Principally from U. S. Navy)	

## The Expenses were made up of the following:

Payments directly to cooperating institutions.....	6,200
Navy projects carried on at the Pittsburgh Laboratory and at Case School of Applied Science.....	19,000
Other projects carried on at Pittsburgh Laboratory.....	3,260
Expenses apportionable to the Laboratory and cooperating institutions....	9,390

Summary reports of the work of the individual Technical Advisory Committees are given as follows: The resurgence of interest shown by these important committees in the research work of the Society is evidenced by the record number of Technical Advisory Committee workings, 17 being held at the present session.

## Technical Advisory Committees

**SENSATIONS OF COMFORT**—Thomas Chester, *Chairman*; N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, F. C. McIntosh, A. B. Newton, B. F. Raber.

Though no laboratory investigations were possible under this Committee during the year, the opportunity was taken to commence the re-study and tabulation of data collected at Minneapolis in 1937. These studies dealt with the general reactions of a large number of office workers to summer cooling and air conditioning and their shock experiences after entering and leaving cooled and air conditioned offices and the general findings were published in the 1938 Transactions.

Some 25,000 cards collected at that time are being analyzed, with the aid of a tabulating machine, under the direction of Professor Rowley at the University of Minnesota. Mr. Newton who conducted the original tests is assisting in the work. The cards are being sorted and correlations made between the more important combinations of the following variables: (a) Feeling of comfort (scale ranges from uncomfortably cold to uncomfortably warm); (b) Variation in feeling of comfort at various periods of the day; (c) Dry-bulb temperature of the conditioned space; (d) Relative humidity of the conditioned space; (e) Effective temperature of the conditioned space; (f) Age of the subject reporting reactions; (g) Sex of the subject reporting reactions; (h) Outdoor dry-bulb temperature; (i) Outdoor relative humidity; (j) Work zone.

From these tabulations correlations will be made between such variables as age and comfort, effective temperature and comfort upon entering the conditioned space, effective temperature and comfort reactions after continued exposure in conditioned space, relative humidity and comfort reactions, etc. The exact course of the project can be determined only after some of the initial investigations have been made.

Some of the studies conducted for the U. S. Navy Department developed data of interest to this Committee but its import cannot, of course, be made available at this time and will await release by the naval authorities.

**PHYSIOLOGICAL REACTIONS**—R. W. Keeton, *Chairman*; Thomas Bedford, A. R. Behnke, A. C. Burton, E. F. DuBois, A. W. Eyer, A. P. Gagge, F. C. Houghten, A. C. Ivy, R. R. Sayers, Charles Sheard, A. D. Tuttle, C.-E. A. Winslow.

Important government research assignments at the College of Medicine of the University of Illinois in Chicago have prevented a full time resumption of the studies which were discontinued in October, 1942, and are not likely to be renewed until conditions are materially changed. A report covering observations on the physiological adjustments of subjects passing from a hot to a comfortable environment was made to the Committee at the end of the year by Dr. Keeton and his colleague, Mr. Glickman. Dr. R. W. Keeton, the Chairman of the Committee, revised Chapter 37 of THE GUIDE dealing with Air Conditioning in the Treatment of Disease.

Numerous studies on the effect of environmental conditions on the physiological reactions of the human being have been and are being made for the Armed Services of all the allied nations. This work, which cannot be made public now, will eventually add tremendously to our knowledge of this subject and it will be the

function of this Committee to interpret the results for the Society membership and the general public. Some of the Committee met informally in Chicago in November and commenced the formulation of plans for studies at the Society's research laboratory and at cooperating institutions. The Committee expressed the opinion that the Society should continue its studies into the physiological reactions of human beings to environmental conditions as soon as conditions allowed and staff were available. The Committee also expressed the need for wider publication of the Society's work particularly in journals read by the medical profession and physiologists.

**REMOVAL OF ATMOSPHERIC IMPURITIES**—R. S. Dill, *Chairman*; J. J. Burke, J. M. Dalla Valle, Leonard Greenburg, Theodore Hatch, W. C. L. Hemeon, L. R. Koller, C. A. McKeeman, F. H. Munkelt, H. C. Murphy, G. W. Penney, E. B. Phelps, F. B. Rowley, G. H. Schember, J. B. Smith, W. O. Vedder, J. H. Waggoner, R. P. Warren, W. F. Wells.

Work has continued under this Committee under a cooperative agreement with the University of Minnesota where Prof. F. B. Rowley and his colleagues have been studying problems connected with the testing of air filters.

A paper entitled, *Discoloration Methods of Rating Air Filters*, is being presented by F. B. Rowley and R. C. Jordan at the 50th Annual Meeting of the Society. In this paper the authors present, in addition to a review of the principal photometric methods used in testing air filters, a new method of determining the true discoloration efficiency of air filters; they also analyze the theory of discoloration. They introduce the term *value* in order to take into consideration the psychological and physiological reactions of the human eye in interpreting discoloration.

In the latter part of the year the air filter research equipment described in previous reports has been relocated and some changes in design made. A special air filter research room has been constructed and improvements incorporated, which have resulted from past experience. Recent investigations were mainly concerned with the air-pressure gradients through filters and it is expected that a paper covering this phase of the work will be presented at the next Semi-Annual meeting of the Society.

**RADIATION AND COMFORT**—J. C. Fitts, *Chairman*; E. L. Broderick, R. E. Daly, L. N. Hunter, F. W. Hutchinson, A. P. Kratz, John James, C. S. Leopold, D. W. Nelson, G. W. Penney, W. R. Rhoton, T. H. Urdahl.

Though exploratory tests to serve as a guide for future research under this committee were planned for the Research Laboratory for 1943, pressure of the Navy work prevented their being made in the winter months. Future work under this Committee is to a marked extent dependent on some satisfactory solution of the problem of instrumentation with regard to the measurement of radiant heat; to this end the Committee is meeting jointly with the Technical Advisory Committee on Instruments, at the 50th Annual Meeting, to plan a laboratory and field attack on this problem.

An important and fundamental investigation was made under this Committee during 1943 at the University of California, Berkeley, Calif., where Profs. B. F. Raber and F. W. Hutchinson are studying the effect of radiant interchange on comfort. Their second progress report on the work is being presented in the form of a paper entitled *Optimum Surface Distribution in Panel Heating and Cooling Systems* (see p. 231).

Work is in progress at California to investigate the relationship between the air temperature and surface temperature in a panel heated room and to establish the magnitude of air temperature depression (below MRT) which is thermally possible in a room heated with low temperature ceiling panels.

**INSTRUMENTS**—D. W. Nelson, *Chairman*; L. M. K. Boelter, E. L. Broderick, R. S. Dill, R. B. Engdahl, A. P. Gagge, J. A. Goff, J. D. Hardy, A. E. Hershey, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

Though no work has been carried out directly under this Committee during the past year, some studies having a close bearing on it were undertaken at the Research Laboratory when the Navy work had been completed, whilst at the University of

Illinois and at the University of California studies on instruments formed part of larger and more comprehensive investigations. Most instrumental development work is carried on as an integral part of some research project for which adequate instruments are not available; it is not generally practicable or easily possible for the instrumentation for a project and the project itself to be separated.

There is a great need for the standardization of both Laboratory and field instruments for the measurement of mean radiant and low air velocities. Many field studies give inconclusive results because of difficulties in the interpretation of the results of some of the measurements made. Considerable attention to this subject is proposed for the Society's Research Laboratory in the near future.

**WEATHER DESIGN CONDITIONS**—T. H. Urdahl, *Chairman*; J. C. Albright, H. S. Birkett, P. D. Close, J. F. Collins, Jr., John Everetts, Jr., C. M. Humphreys, H. H. Koster, J. W. O'Neill, F. W. Reichelderfer.

It has not been possible to resume actively the work of this Committee during the past year. The subject has, however, become of considerable interest in connection with fuel saving and the data which are now being collected in many parts of the country are much more complete than was collected before the war. It is intended during the coming year to commence work on the preparation of weather design data of use to the design engineer by an analysis of data collected some years ago and on which a preliminary analysis was done under governmental agencies.

**RADIATION WITH GRAVITY AIR CIRCULATION**—M. K. Fahnestock, *Chairman*; R. E. Daly, R. S. Dill, A. G. Dixon, F. E. Giesecke, H. F. Hutzler, J. P. Magos, J. W. McElgin, J. F. McIntire, W. A. Rowe.

During the past year the active work of this Committee was carried on in the warm wall test booths in the Engineering Laboratories of the University of Illinois. The work included the following studies, each made with one small-tube type of direct cast-iron radiator, one cast-iron convactor with end and bottom connections, one non-ferrous convactor with end connections, and one non-ferrous convactor with bottom connections.

1. Heat Output Studies With: (a) Water at different velocities or temperature drops through the units; (b) Water at different entering and mean temperatures; (c) Steam and water at the same mean temperatures.

2. Fluid Pressure Loss or Frictional Resistance Studies With: (a) Top and bottom end connections to the radiator; (b) End and bottom connections to the cast-iron convactor; (c) End connections to one non-ferrous convactor and bottom connections to another non-ferrous convactor. (All units were tested with the connections made of two sizes of pipe.)

The work was made possible by obtaining the temporary services of Dr. F. E. Giesecke, who is on retirement from the Agricultural and Mechanical College of Texas. A preliminary report covering a portion of this work and entitled, Friction Heads in Radiators, Convectors, and Pipes, by Dr. F. E. Giesecke was given to the members of the Committee under date of March 2, 1943. The results of the completed work are available for a technical paper to be presented at the Summer Meeting of the Society in 1944.

**HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION**—W. E. Heibel, *Chairman*; William Goodman, H. F. Hutzler, Ferdinand Jehle, S. F. Nicoll, R. H. Norris, L. P. Saunders, R. J. Tenkonohy, G. L. Tuve, D. C. Wiley.

A cooperative experimental program is being started at Case School of Applied Science for the study of evaporating Freon 12 refrigerant transfer inside horizontal tubes. It is hoped to enlarge this program later to take full advantage of the experience gained by the School in the Navy and other coil test programs. Among proposed future subjects of study are:

1. Heat transfer and air resistance of finned coils condensing moisture from the air.
2. The effect of fin and tube proportions on heat transfer condensed moisture carry-over and air resistance.

3. Correlation of experimental and theoretical fin heat flow resistance.
4. Preferred method of coil performance correlation and prediction.

**COOLING LOAD IN SUMMER AIR CONDITIONING**—W. E. Zieber, *Chairman*; John Everetts, Jr., W. F. Friend, R. H. Heilman, John James, C. S. Leopold; C. O. Mackey, A. E. Stacey, Jr., G. E. Tuckerman, J. H. Walker, M. J. Wilson.

A cooperative project under the direction of the Committee was initiated this year at Cornell University for the study by mathematical methods of the heat flow through walls due to temperature difference and sunlight. The present study takes advantage of mathematical methods previously developed under the sponsorship of the Pierce Foundation and reported at the 1943 Annual Meeting in a paper by C. O. Mackey and L. T. Wright, Jr.<sup>1</sup> The study is a continuation of the long-term experimental program which has been carried on at the Laboratory. It should result in a better knowledge of this important subject through better control of the many uncontrolled and unknown variables encountered in the experimental program.

The study this year dealt primarily with homogeneous walls. It is hoped to continue with the study of non-homogeneous walls and also to determine the storage effect of building mass with changing inside temperature. A progress report covering the work done this year has been presented to the Committee and is expected to result in a paper for the Semi-Annual Meeting.

Another phase of this project is the study of the normal summer air temperatures and solar radiation on which a paper is also expected for the Semi-Annual Meeting.

During the year a sub-committee was appointed consisting of John Everetts, Jr., *Chairman*, A. E. Stacey, Jr., and J. H. Walker to study and correlate the results of the experimental and mathematical work carried on under the auspices of the Committee on roofs and walls.

Study is also urgently needed on the subjects:

1. Infiltration through opened, swinging and revolving doors.
2. Heat load from appliances.
3. Heat load from lighting.

**AIR DISTRIBUTION AND AIR FRICTION**—J. H. Van Alsbury, *Chairman*; S. H. Downs, A. P. Kratz, E. C. Lloyd, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, L. P. Saunders, M. C. Stuart, Ernest Szekeley, G. L. Tuve.

Though the activities of this Committee have been reduced and many of the research projects previously carried out under its direction have been temporarily discontinued, it was found possible to conduct one project at Case School of Applied Science where results of tests with about 45 sizes and kinds of non-directional discharge outlets were correlated with data from other investigations. This work was reported in a paper presented at the 50th Annual Meeting of the Society by G. L. Tuve and G. B. Priester and was entitled, *The Control of Air Streams in Large Spaces* (see p. 153).

**HEAT REQUIREMENTS OF BUILDINGS**—P. D. Close, *Chairman*; E. K. Campbell, J. F. Collins, Jr., E. F. Dawson, W. H. Driscoll, H. H. Mather, H. K. McCain, M. W. McRae, C. H. Pesterfield, F. B. Rowley, R. K. Thulman.

The basement heat loss study in the simulated basement on the premises of the Bureau of Mines at Pittsburgh was continued during the year, adding to the data presented in the original paper,<sup>2</sup> presented in 1942. Some additional data were obtained on the effect of insulation on heat loss in winter and condensation in summer. Unfortunately the set-up and the ground around it was unavoidably disturbed, thus largely nullifying the value of the later tests. It is believed that additional data of

<sup>1</sup> Summer Comfort Factors as Influenced by Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 148.)

<sup>2</sup> A.S.H.V.E. RESEARCH REPORT NO. 1213—Heat Loss through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 369.)

value might be obtained by extending the studies to actual basements and ground floors.

During the year the Committee gave consideration to the problem of exposure factors and found a wide range of opinions as to their value. Prevailing opinion held them unnecessary to increase overall estimated heat loss but of some value in balancing heat supplied to exposed and shielded rooms.

Problems of interest for future study appear to be:

1. Over-all infiltration, particularly when considering type of heating system, fireplaces used, etc. Interest arises out of the wide diversity of opinion as to the proper basis for estimating infiltration.
2. Reconciliation of building heat loss with seasonal heating requirements.

Two papers relating to the activities of the Committee have been prepared by the Chairman, one was presented<sup>3</sup> at the summer meeting and the other is selecting winter design temperatures (see p. 281).

**AIR CONDITIONING REQUIREMENTS OF GLASS**—R. A. Miller, *Chairman*; L. T. Avery, F. L. Bishop, W. A. Danielson, H. C. Dickinson, J. D. Edwards, J. E. Frazier, E. H. Hobbie, C. L. Kribs, Jr., Axel Marin, F. W. Parkinson, W. C. Randall, L. E. Seeley, L. T. Sherwood, J. T. Staples, H. B. Vincent, G. B. Watkins, F. C. Weinert.

Since the Committee decided not to attempt any detailed research program during the war unless a special need arose, it has not been very active during 1943. It is expected to become active in 1944 since some earmarked funds are available for studies contemplated with the idea of obtaining additional data on the subject of window shades to satisfy the questions which have been raised about the present data.

The subject of infiltration is one which has attracted considerable attention in the last year or so because of government publicity in connection with methods of fuel saving. It is of some interest to this Committee and a program of research is contemplated.

**SOUND CONTROL**—R. D. Madison, *Chairman*; W. W. Kennedy, V. O. Knudsen, C. H. Randolph, A. E. Stacey, Jr., A. G. Sutcliffe, T. A. Walters, R. M. Watt, Jr.

The cooperative experimental program for the study of sound attenuation in ducts, elbows, outlets and splitters which was started at Rensselaer Polytechnic Institute in 1940 was at a standstill during the year. It is hoped to conclude it after the war. In addition the Committee is considering suggestions for study of:

1. Generation of noise by mechanical equipment and its measurement.
2. Room noise level problems including the introduction of noise through outlets, walls and floors, the direct generation of noise by equipment in the room and the effect of the characteristics of the room on the noise.
3. Desirable noise level limits including a study of the annoying effect of certain types of noise.

**COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS**—H. B. Nottage, *Chairman*; C. F. Boester, W. W. Cockins, S. C. Coey, E. H. Kendall, E. R. Ketchum, S. R. Lewis, J. F. Park, S. I. Rottmayer, E. W. Simons, E. H. Taze.

The past year has seen no basic changes in either the scope or direction of the cooperative research program under way at the University of California. The experimental program has been concerned with the announced current objective of obtaining performance and design data on point-to-point conditions within a slot-packed, mechanically-induced draft cooling tower.

Based upon experimental work conducted during 1941-1942, a paper on spray nozzle performance<sup>4</sup> was presented at the Semi-Annual Meeting of the Society at Pittsburgh, in June, 1943, by L. M. K. Boelter and S. Hori. This was the second in a series dealing with spray tower performance.

<sup>3</sup> Graphical Method of Calculating Heat Losses, by Paul D. Close. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 345.)

<sup>4</sup> A.S.H.V.E. RESEARCH REPORT No. 1240—Spray Nozzle Performance in a Cooling Tower, by L. M. K. Boelter and S. Hori. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 309.)

Progress on the experimental program at the University of California has been summarized in a special illustrated report prepared under the direction of Professor Boelter. (A copy is available on loan to specially interested parties.) This report deals primarily with the development of transfer conditions throughout all parts of the full-scale experimental induced-draft cooling tower. These instruments represent essential original contributions to cooling tower studies. Preliminary test data have shown their value in the analysis and understanding of conditions within a cooling tower.

Further analysis of the spray tower data is being undertaken at the University of California.

The Chairman undertook the review of Chapter 27 of THE GUIDE, dealing with Spray Equipment.

**PSYCHROMETRY**—J. A. Goff, *Chairman*; F. R. Bichowsky, W. H. Carrier, H. C. Dickinson, R. S. Dill, A. W. Gauger, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, A. P. Kratz, D. M. Little, Axel Marin, D. W. Nelson, W. M. Sawdon.

The activities of the Committee for the past year have been concerned with the revision of Table 6 of THE GUIDE. To that end a paper<sup>5</sup> was presented at the Semi-Annual Meeting of the Society in Pittsburgh in June 1943. This paper reports the experimental results of the cooperative investigation sponsored by the Society at the Towne Scientific School, University of Pennsylvania.

At the Semi-Annual Meeting of the Society the Committee held a meeting in which it was decided to recommend to the Guide Publication Committee that the proposed revision of Table 6 for the range  $-100$  to  $+200$  F be accepted for publication.

The proposed revision of Table 6 has not yet been completed, since it was found that available data on  $B_{wv}$  (the second virial coefficient of water) were not sufficiently reliable, necessitating a careful review of the literature, in particular the low pressure steam properties published by the Bureau of Standards, 1939, in the light of modern statistical mechanical theory. This review is not yet complete but it seems that  $B_{wv}$  can be inferred from the above data with sufficient reliability.

At the meeting of the Committee on June 7 there was some discussion as to the proper graphical representation of the data of Table 6 and those discussions have continued between members of the Committee.

The future activities of the Committee are immediately concerned with completion of the revision of Table 6, and with the possibility of preparing tables similar to Table 6 but for pressure other than atmospheric.

New knowledge of the thermo-dynamic properties of moist air which it is expected will be characterized by a high degree of accuracy will, it is felt, emphasize the need for the development of a suitable dew-point or other apparatus having considerable accuracy.

**FLOW OF FLUIDS THROUGH PIPES AND FITTINGS**—F. E. Giesecke, *Chairman*; T. M. Dugan, S. R. Lewis, L. P. Saunders.

During the year the Chairman completed the manuscript for a bulletin, to be issued through the Engineering Experiment Station of Texas Agricultural and Mechanical College, entitled, Friction Heads in Six-Inch Pipe and the Effects of Pipe Surface Roughness and of Temperature on Friction Heads. This bulletin contains, among other data, that published<sup>6</sup> in the A.S.H.V.E. Journal Section in November 1942.

For future research the following problems have been suggested:

1. How does thermal circulation of water take place in a vertical pipe?
2. To what extent is the friction in a stream of water flowing in a pipe affected by a turbulence created at the end of the pipe or at some point in the pipe? For example, by a pipe fitting, or a valve, or an orifice, or a pump.

<sup>5</sup> Research Report No. 1238—Final Values of The Interaction Constant for Moist Air, by John A. Goff, J. R. Anderson and S. Gratch. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 269.)

<sup>6</sup> Friction Heads Due to Water Flow in Copper, Brass and Other Smooth Pipes, by F. E. Giesecke. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 175.)

3. To what extent is the friction of a stream of water flowing in a pipe affected by the transfer of heat from the stream of water to the enclosing pipe wall or from the enclosing pipe wall to the stream of water?

FUELS—R. A. Sherman, *Chairman*; R. M. Conner, R. S. Dill, R. B. Engdahl, A. C. Fieldner, L. N. Hunter, S. Konzo, W. M. Myler, Jr., H. J. Rose, C. E. Shaffer, T. H. Smoot, R. K. Thulman, T. H. Urdahl, E. C. Webb.

This Committee has been badly hampered in carrying out its projected program, drawn up at the 49th Annual Meeting of the Society because of the lack of competent personnel that could be spared from other duties, mostly in connection with the Navy studies at the Research Laboratory. Work on a projected program on chimneys had to be postponed to a more favorable time but it is hoped that an active program will be in operation in the near future. A ballot was conducted among members of this Committee late in the year in an effort to select projects to be recommended to the Research Committee for investigation. Among the projects calling for early attention were the following:

1. Fundamental and comprehensive investigation of factors governing the performance of chimneys.
2. Investigation of the performance of barometric dampers on residential heating equipment.
3. Methods for the measurement of flue-gas temperatures in heating equipment.
4. Methods for the measurement of the temperature of metal and other surfaces in heating equipment.
5. Investigate the relation of the heating requirement of buildings and the required boiler output.

Bituminous Coal Research Inc. has on its project agenda an investigation of chimneys and it is expected that cooperation will be arranged on this project.

In December a preliminary report was made to the Committee by a sub-committee set up to prepare criteria for fuel burning heating equipment. The report served to point out the fundamentals of the problem, the need for close cooperation between the heating industry and the insulation industry in achieving a satisfactory thermal environment, and the need for the standardization of methods of calculating heat losses from a structure, particularly as regards infiltration losses.

One of the members of the Committee, R. S. Dill, was responsible for revising Chapter 9 of THE GUIDE, which deals with chimneys and draft calculations. Another member, S. Konzo, was also a member of the Guide Publication Committee.

CORROSION—L. F. Collins, *Chairman*; R. C. Doremus, E. W. Guernsey, G. G. Marvin, A. R. Mumford, L. P. Saunders, F. N. Speller.

Officially, the research studies being conducted under a cooperative agreement at Carnegie Institute of Technology, Pittsburgh, were concluded in September. These studies were aimed at revealing the mechanism by which carbon dioxide entrained by steam becomes dissolved in the condensate formed in heating equipment. Because of obstacles imposed by the war, the work consumed much more time than was contemplated. In September, Dr. McKinney and his co-workers were not entirely satisfied with the results and elected to carry on at their own expense for an additional period. They have now completed the work and are preparing a report which will be submitted to the Committee at the 50th Annual Meeting. It is felt that this work has produced some usable results.

The Chairman presented a paper before the Fourth Annual Water Conference of the Engineers Society of Western Pennsylvania, held at Pittsburgh, in November, entitled, More Information Concerning Corrosion in Steam Heating Systems.

The Committee has been asked to consider an extension of the scope of its program, which has heretofore been mainly concerned with corrosion in steam heating systems, so as to cover the extremely divergent corrosion problems encountered by the many branches of the air conditioning industry.

AIR CONDITIONING IN INDUSTRY—W. L. Fleisher, *Chairman*; L. T. Avery, A. R. Behnke, Leonard Greenburg, W. E. Heibel, F. C. Houghten, D. E. Humphrey, E. F. Hyde, L. L. Lewis, P. A. McKittrick, R. R. Sayers, R. M. Watt, Jr., H. E. Ziel.

Though this Committee laid plans for an extensive program of laboratory and field studies in various industries where hot and humid conditions are encountered,

lack of trained personnel and funds prevented the program from being carried out during the past year.

The Committee is meeting in New York at the 50th Annual Meeting to consider:

1. The examination and analysis of existing laws and ordinances as issued by various organizations interested in industrial hygiene.
2. The re-examination of laboratory data on the effect of hot environments on man.
3. The collection of data in the field on the effect of environmental conditions on output and fatigue of workers.

**SORBENTS**—F. R. Bichowsky, *Chairman*; O. D. Colvin, F. C. Dehler, John Everetts, Jr., Ralph Fehr, J. A. Goff, W. R. Hainsworth, C. H. B. Hotchkiss, J. C. Patterson, G. L. Simpson.

This Committee is still somewhat new and without a fully organized program. It was formed to correlate and extend the data concerning both adsorbent and absorbent materials, processes and utilization. At a meeting coincident with the 1944 Annual Meeting of the Society it is hoped to develop a specific program along the lines of:

1. Chemical and physical properties and characteristics of sorbents.
2. Equipment and methods of application.
3. The fields of economical application of sorbents.

**INSULATION**—E. R. Queer, *Chairman*; J. D. Edwards, F. G. Hechler, H. K. McCain, Paul McDermott, W. T. Miller, H. C. Robinson, F. B. Rowley, G. L. Tuve, J. H. Waggoner, G. B. Wilkes.

The committee accomplished the following work:

1. A review was made of data from various sources bearing on the subject of upward heat flow in mineral wool insulation and the possible effect of convection currents on such heat flow. The conclusions of the sub-committee were that the available data and information do not indicate that any change in the conductivity value of 0.27 for mineral wool as now listed in *THE GUIDE*, 1943, is necessary or desirable for estimating purposes. The effect of any wooden joists or other building members in such a position as to affect the flow of heat through the insulating material should be taken into account. One of the most important phases of independent tests, with and without paper over the exposed surface of the mineral wool, showed no effect due to the paper covering that was not within the limits of experimental error.

2. It is the opinion of this Committee that it is desirable to have insulation heat transmission coefficients under actual service conditions. The Office of Production, Research and Development (OPRD) of the WPB has an extensive research and testing program to determine the heat transmission performance under accelerated climatic conditions of materials used in economical housing. An arrangement has been worked out whereby these data will be made available to this committee with the thought of eventually using them in *THE GUIDE*.

3. A sub-committee of ASTM C-16 under the chairmanship of R. H. Heilman is studying vapor barriers and the mechanism of vapor transmission. This committee as well as a member of the ASRE is cooperating in this study.

The additional work to be undertaken under this committee is:

1. A review of insulation coefficients in *THE GUIDE* so as to bring them up to date. If found necessary tests should be made by the new standardized Guarded Hot Plate Test Method. A Standardized Guarded Hot Box Test Method should be developed for overall coefficients by the combined A.S.H.V.E., ASTM and ASRE committees.
2. Low temperature heat transmission values should be obtained for 10 in. and 12 in. thick walls.

**HEAVY DUTY AIR FURNACES**—E. K. Campbell, *Chairman*; H. D. Campbell, K. T. Davis, A. P. Kratz, W. J. MaGill, A. A. Olson, B. B. Reilly, H. J. Rose, H. A. Soper.

This committee was set up in the latter part of 1942 to analyze information on heavy duty furnaces with a view to determining whether sufficient data were available and whether a code committee was needed. The Committee reported in June that although there are certain codes or tentative codes for furnaces of smaller output there is, at present, no generally recognized authority sponsoring a code covering the general field of heavy duty air heating furnaces. They therefore recommended that the A.S.H.V.E. is the proper authority and that the Society should develop and adopt, for the public benefit, a code for testing and rating heavy duty air heating furnaces.

At the November meeting of the Council held in Indianapolis, authority was given for the appointment of a Code Committee.

ATLANTA CHAPTER ADVISORY COMMITTEE—T. T. Tucker, *Chairman*; W. J. McKinney, Leo Sudderth.

#### *Attic Fans*

Work originally commenced in 1941 under a cooperative agreement between the Society and the Georgia School of Technology resulted in a paper given at the 48th Annual Meeting of the Society held in January 1942. This contract was renewed on a reduced scale in 1942, and was again renewed in 1943 without payment of any funds since the institution found itself unable to proceed at present with further experimental work.

A report covering the work completed in 1942 has been prepared by Professor Hinton and Mr. Wanamaker and approved by the Atlanta Chapter Committee (see p. 371).

OREGON CHAPTER RESEARCH ADVISORY COMMITTEE—E. W. Neubauer, *Chairman*; J. E. Yates, W. J. Kollas, J. D. Kroeker, T. E. Taylor, E. C. Willey.

*Heat Flow Through Wet Building Walls:* Late in 1942 a cooperative agreement was signed between the Society and Oregon State College, Corvallis, Ore., covering the investigation of heat flow through wet building walls to be carried on in the domestic heating laboratory of the College. The study was planned jointly by Prof. Earl C. Willey of the College and the research committee of the Oregon Chapter of the Society of which E. W. Neubauer of Portland is chairman. The plans called for the construction of various types of building wall sections and their testing under conditions simulating heat loss during rainy weather.

It was felt that the results of the investigation would be of particular interest for application in the Pacific Northwest in which building walls are wet during much of each winter season, since only incomplete information on heat loss through wet walls is available for accurate determination of heating plant requirements.

The contract was renewed in June 1943 for a further year. During the first year the work consisted mainly in reconstructing the original apparatus to take advantage of the information gained by the University of Minnesota Experiment Station in the operation of a hot box similar to the one to be used in these studies.

Like other institutions Oregon State College found itself too short of competent help to proceed as rapidly as had been planned and there is little likelihood of completing this project within the period covered by the contract. Sufficient data have been collected however to indicate the trend of the investigation which will be resumed at a more rapid tempo as soon as conditions permit.

#### Research Papers—1943

1. The Performance of Side Outlets on Horizontal Ducts, by D. W. Nelson and G. E. Smedberg (Wisconsin) (Research Report No. 1226, A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 58).
2. Spray Nozzle Performance in a Cooling Tower, by L. M. K. Boelter and S. Hori (California) (Research Report No. 1240, A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 309).
3. Final Values of the Interaction Constant for Moist Air, by John A. Goff, J. R. Anderson and S. Gratch (Pennsylvania) (Research Report No. 1238, A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 269).
4. Discoloration Methods of Rating Air Filters, by Frank B. Rowley and Richard C. Jordan (Minnesota) (See p. 173).
5. Optimum Surface Distribution in Panel Heating and Cooling Systems, by B. F. Raber and F. W. Hutchinson (California) (See p. 231).
6. Control of Air-Streams in Large Spaces, by G. L. Tuve and G. B. Priestler (Case) (See p. 153).
7. Physiological Reactions Applicable to Workers in Hot Industries, by F. C. Houghten, Carl Gutberlet and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 188). (Research Laboratory.)
8. Progress in Development of Standards for Comfort Air Conditioning, by Lt. Comdr. F. C. Houghten (See p. 87).

## Institutions Cooperating with the Committee on Research

*Carnegie Institute of Technology:* Corrosion in steam heating systems. *Case School of Applied Science:* Air distribution in air conditioned spaces. *Cornell University:* Heat flow through building walls. *Georgia School of Technology:* Cooling of a structure with attic fans. *Oregon State College:* Heat transfer through wetted walls. *University of California:* Performance of Cooling Towers; Radiant Heating and Cooling. *University of Minnesota:* Methods of testing air cleaning devices; Statistical study of reactions of office workers to environmental conditions. *University of Pennsylvania:* Measuring departures from Dalton's Law of Air-Water vapor mixtures.

Contracts at the following institutions were in force, but were inactive during 1943: *Agricultural and Mechanical College of Texas, Lehigh University, Rensselaer Polytechnic Institute, University of Illinois* (College of Engineering and Medical School), and the *University of Wisconsin*.

## CONTRIBUTORS TO RESEARCH

## Financial Contributions

Aerofin Corp., Air-Maze Corp., The Harry Alter Co., Aluminum Co. of America, Auditorium Conditioning Co., Automatic Burner Corp., Automatic Products Co., Barber-Colman Co., Barnes & Jones, Inc., Bayley Blower Co., Bell & Gossett Co., Blue Ridge Glass Corp., Buffalo Forge Co., California Redwood Association, E. K. Campbell Heating Co., Cutler-Hammer, Inc., Chase Brass & Copper Co., Inc., Chicago Pump Co., Chamberlin Metal Weather Strip Co., Crane Co., Calgon, Inc., Carrier Corp., Drayer & Hanson, Inc., The Dole Valve Co., Duquesne Light Co., Detroit Stamping Co., Lewis M. Ellison (In Memory of Albert O. Ellison), Lewis M. Ellison, M. A. Gerrett Corp., The G & O Mfg. Co., Grant Wilson, Inc., Grinnell Co., Inc.

The Fluor Corp., Ltd., Fedders Mfg. Co., Inc., The Fulton Sylphon Co., Forslund Pump & Machinery Co., The Heil Co., Heating, Piping and Air Conditioning Contractors National Association, Inland Steel Co., Illinois Engineering Co., Insulation Board Institute, Johnson Service Co., Johns-Manville, Kieley & Mueller, Inc., Monarch Mfg. Works, Inc., The Marley Co., Inc., Masonite Corp., May Oil Burner Corp., Modine Mfg. Co., John J. Nesbitt, Inc., Narowetz Heating and Ventilating Co., The Nash Engineering Co., National Lumber Manufacturers Association, The Plastergong Wall Board Co., A. G. Pratt, The Permutit Co., Perfex Corp., Portland Cement Association, Penn Lithographing Co., Pipe Fabrication Institute.

Surface Combustion, Sheffler-Gross Co., The Sisalkraft Co., Spencer Thermostat Co., Subsidiary Companies of Bethlehem Steel Corp., Trade-Wind Motorfans, Inc., The Torrington Mfg. Co., Tempil Corp., The Timken-Detroit Axle Co., The Trane Co., Utility Fan Corp., U. S. Electrical Motors, Inc., Universal Power Corp., U. S. Gauge Co., Union Electric Co. of Missouri, Universal Cooler Corp., The Vilter Mfg. Co., Watson & McDaniel Co., Williams Oil-O-Matic Heating Corp., Webster Electric Co., Weil-McLain Co., L. J. Wing Mfg. Co., Westinghouse Electric & Manufacturing Co., Wolverine Tube Division, Wright-Austin Co., York Ice Machinery Corp.

## Periodicals

*American Artisan, Combustion, Domestic Engineering, Fuel Oil and Oil Heat, Heating, Piping and Air Conditioning, Heating and Ventilating, Ice and Refrigeration, and the Journal of the Institution of Heating and Ventilating Engineers* (London).

## New Research Laboratory

On March 15, 1944, the Society's Research Laboratory, which had been located for the past 25 years in the building of the U. S. Bureau of Mines, Pittsburgh, Pa., was moved to 10700 Euclid Ave., Cleveland 6, Ohio.

The building is located at the corner of East 107th Street and Euclid Avenue and is a short distance from both the Euclid Avenue Station of the Pennsylvania Railroad and the East Cleveland Station of the New York Central System.

With exterior dimensions of 91 ft by 57 ft, the building has two main floor sections, a front and a rear mezzanine, and a large basement. The major portion of both the main and third floors measures 51 ft by 33 ft deep and has a ceiling height of over 20 ft. Some 2200 sq ft of basement space have a

10 ft ceiling height. Office space is available on all floors. The building is heated by district steam.

The whole building will be available to the Society for its research work and, as announced at the 50th Annual Meeting of the Society, a two-year lease has been signed with the owners on terms considered favorable to the Society.

The removal to this building offers a challenge to the whole membership of the Society and to the Industry they represent to expand the research work and ensure that, as Research has played an important part in *the Great Past* of the Society, it will play a still more important part in the Society's *Greater Future*.

E. K. Campbell, Kansas City, Mo., presented the Report of the Treasurer, which was received and filed, followed by the Report of the Finance Committee by J. F. Collins, Jr., *Chairman*.

### Report of the Treasurer

It is my privilege at this time to make a very favorable and pleasing report on the condition of our Society treasury. It is my privilege for the reason that, while the treasurer is in the limelight, the chairman of the Finance Committee does the work. I can say to you that the Finance Committee has done fine work during the past year. J. F. Collins, Jr., as chairman, with the approval of the balance of the committee, has changed and improved the set-up of the bookkeeping, has changed and improved the arrangement with the auditors, all of this being in connection with the change in the fiscal year. This change in the fiscal year affects no member, but makes it easier for the Council to have correct information for its January meeting. Mr. Collins has simplified the entire statement and made it more easily understandable.

Your president, in his report, gave most of the figures, and I can only add the statement that all of the investments of the Society are now in U. S. Government securities, or in securities of the Government of Canada. I may explain that, because of the adverse exchange rates, the Canadian members' dues are deposited in a Canadian bank.

The management of your Society has been good; the funds have been carefully watched; and now, the danger which befalls any Society like ours, comes, as it does to an individual. The average individual can stand adversity much better than he can stand prosperity. When money is easy, the temptation to spend grows. And so, I warn you that, in our Society, there will come proposals to spend money for this, or that, or some other idea, which may or may not be worthy in itself, but which might easily be too much of a drain on the national treasury at times when it is hard for members to pay their dues, and when the Society loses members by the hundreds on that account. There was a time when our membership dropped from around 1800 to about 1200 because of the depression. We can expect such times again, and our expenditures must be set up in such a way that they can be reduced as necessity requires.

The various proposals which are likely to come for the spending of money remind me of two women discussing their husbands. One told how wonderfully generous her husband was—if she wanted a new car, or a new coat, all she had to do was to mention it. She went on to some length until her friend interrupted with, "Well, does he give you all the money you want to spend?" The reply was instantaneous—*there ain't that much*. So, when these proposals come to raid the national treasury for this or that, just remember the reply *there ain't that much*, because the time will come in the next depression—maybe 10 or 15 years from now—when there will not be that much. The Society can be wrecked by improper and too liberal management between now and then.

Respectfully submitted,

E. K. CAMPBELL, *Treasurer*

**Accountants' Report****TUSA & LABELLA**CERTIFIED PUBLIC ACCOUNTANTS  
52 WILLIAM ST., NEW YORKAmerican Society of Heating and  
Ventilating Engineers,  
51 Madison Avenue,  
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the related Funds for the period from January 1, 1943, to October 31, 1943, and submit herewith our report.

The audit covered a verification of the assets and liabilities as of the close of business October 31, 1943, and a review of the operating accounts for the period then ended. For the period audited the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected, compared with the cash records and supported by payment vouchers; also the dues income and interest income from savings accounts and securities were accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1943, is submitted herewith and your attention is directed to the following comments thereon:

**CASH**

Cash on Deposit as reflected in the attached cash schedule was verified by direct communication with the banks and the balances reported to us were reconciled with those on the books of the Society. The petty cash was verified by count.

**MARKETABLE SECURITIES**

The securities, shown on the subjoined schedule, were verified by direct communication with the Bankers Trust Company, where same are deposited for safekeeping. This asset has been included in the Balance Sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

**CERTIFICATE OF INDEBTEDNESS.**

The certificate of indebtedness issued to the Society by Farrar & Trefts, Inc., was verified by inspection of the instrument.

**ACCOUNTS RECEIVABLE**

A list of the membership dues receivable as of October 31, 1943, furnished to us by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. These unpaid dues were aged and summarized as follows:

Dues invoiced during 1943.....	\$6,993.00
Dues invoiced during 1942.....	1,740.00
Dues invoiced during Prior Years.....	361.50
<b>Total.....</b>	<b>\$9,094.50</b>

Amounts due from Guide advertisers and other debtors were verified by trial balance of the individual ledger accounts and found in agreement with the General Ledger Control.

The reserve for dues and sundry accounts receivable found on the books of the Society are ample to cover losses that might result from uncollectible accounts.

## INVENTORIES

The emblems on hand were counted by us, and the inventories of paper and TRANSACTIONS were verified by communication with printers.

These inventories were priced and computed by us. The following TRANSACTIONS were reported to us.

Volume	Year	Quantity	Price	Amount
Prior	Prior	1350	\$1.00	\$1,350.00
44	1938	26	1.58	41.08
45	1939	122	1.66	202.52
46	1940	79	1.25	98.75
47	1941	85	1.32	112.10
47	1941	150 (unbound)	.96	144.00
48	1942	90	1.42	127.80
48	1942	150 (unbound)	1.07	160.50

Total \$2,236.75

## PREPAID TRAVELING

There were on hand railroad scrip books having a value of \$24.46 which are to be turned in for refund.

## PERMANENT ASSETS

Furniture, fixtures and library are shown herein at the book values without appraisal by us; we did, however, provide for depreciation of furniture and fixtures at the rate of ten per cent per annum.

## DEFERRED CHARGES

During the current fiscal period disbursements were made towards the editing and promotion of THE 1944 GUIDE as itemized in the Guide budget which we have deferred to the Guide operations of the ensuing fiscal year.

## ACCOUNTS PAYABLE

All purchase invoices found on file that were applicable to the operations of the current fiscal period were listed by us and the proper liability therefor has been reflected in the attached balance sheet.

In addition, there is due the Research Fund on dues collected from members and associates during the current period the sum of \$994.96 and to the Reserve Fund the sum of \$456.25 for initiation fees collected in the month of October, 1943, plus interest on Treasury Bonds.

## DEFERRED INCOME

Trial balances taken of the Dues Receivable disclosed prepayments made by members and candidates for membership. We have shown these on the subjoined balance sheet as deferred income.

## RESERVE FOR PUBLICATIONS

In accordance with the provision made in the 1943 budget, we have included in the attached balance sheet a reserve of \$3,400.00 to cover the publication of the 1943 TRANSACTIONS—Volume 49.

## FUNDS

An analysis of the following funds reflecting the changes that occurred in these accounts during the period from January 1, 1943, to October 31, 1943, is included herein:

General Fund  
Reserve Fund  
Endowment Fund  
F. Paul Anderson Fund

## 34 TRANSACTIONS AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

There is included herein a complete financial report as prepared for the Committee on Research, setting forth the financial position of the Research Laboratories as of the close of business, October 31, 1943, and the results from operations for the period from January 1, 1943, to October 31, 1943.

Respectfully submitted,  
TUSA & LABELLA,  
Certified Public Accountants.

Dated December 31, 1943.

## BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—  
NEW YORK, N. Y.

(October 31, 1943)

## ASSETS

## GENERAL FUND

## CASH

On deposit.....	\$24,629.82		
On hand.....	100.00	\$24,729.82	
In closed bank.....		380.89	\$25,110.71

## INVESTMENTS (AT COST)

Securities (Market Value \$20,700.00).....		19,900.00	
Add: Accumulated interest...	800.00		
Add: Accrued interest.....	75.00	875.00	20,775.00

## CERTIFICATE OF INDEBTEDNESS

Farrar & Trefts, Inc.....			153.20
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## ACCOUNTS RECEIVABLE

Membership dues.....		9,094.50	
Less: 40% for Research.....	3,418.20		
Less: Reserve for doubtful...	873.84	4,292.04	
		4,802.46	
Advertisers and sundry debtors	1,485.67		
Less: Reserve for doubtful...	962.66	523.01	5,325.47

## INVENTORIES

TRANSACTIONS—Copies.....		2,236.75	
TRANSACTIONS—Paper.....		283.90	
GUIDE—Paper.....		2,561.11	
Emblems.....		58.75	5,140.51

## PREPAID TRAVELING

Railroad scrip.....			24.46
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## PERMANENT

Library.....		300.00	
Furniture and fixtures.....	2,744.14		
Less: Reserve for depreciation	1,489.86	1,254.28	1,554.28

## DEFERRED CHARGES

Guide costs.....	5,754.97		
Prepaid H.P.A.C. subscriptions.....	966.30	6,721.27	\$ 64,804.90

## RESERVE FUND

Cash on deposit.....			1,913.89	
Due from General Fund.....			456.25	
Securities at cost (Market value \$52,241.82).....		\$ 50,719.06		
Add: Accumulated interest....	\$ 1,272.50			
Accrued interest.....	33.75	1,306.25	52,025.31	54,395.45

## ENDOWMENT FUND

Cash on deposit.....		2,255.11		
On hand for deposit.....		15.00	2,270.11	
Securities at cost (Market value \$22,883.70).....		25,153.65		
Add: Accumulated interest....	467.50			
Accrued interest.....	120.00	587.50	25,741.15	28,011.26

## F. PAUL ANDERSON FUND

Cash on deposit.....				1,098.06
				<u>\$148,309.67</u>

## LIABILITIES AND FUNDS

## GENERAL FUND

## LIABILITIES

ACCOUNTS PAYABLE.....	\$ 1,918.72	
TAXES WITHHELD.....	274.91	
ACCRUED ACCOUNTS		
Additional compensation to employees.....	3,654.23	
DUE TO RESEARCH.....	994.96	
DUE TO RESERVE FUND.....	456.25	

## DEFERRED INCOME

Prepaid membership dues..	\$ 570.20	
Less: 40% prepaid to re-search.....	176.20	\$ 394.00

Dues prepaid by candidates for membership.....	283.63	677.63
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## RESERVE FOR PUBLICATIONS

Transactions (1943) Volume 49.....	3,400.00	
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TOTAL LIABILITIES.....	\$11,376.70	
GENERAL FUND.....	53,428.20	\$ 64,804.90

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to an forming a part of this report.

## RESERVE FUND

Principal.....	53,006.10	
Unexpended income.....	1,389.35	54,395.45

## ENDOWMENT FUND

Principal.....	27,432.99	
Unexpended income.....	578.27	28,011.26

## F. PAUL ANDERSON FUND

Principal.....	1,085.83	
Unexpended income.....	12.23	1,098.06

\$148,309.67

## STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS  
NEW YORK, N. Y.*(For the Period from January 1, 1943 to October 31, 1943)*

## INCOME

## FROM MEMBERS

## DUES—RENEWALS

Members and Associates.....	\$45,423.00		
Less: Cancellations.....	1,641.00	\$43,782.00	
Less: 40% to Research Laboratory.....		17,512.80	\$26,269.20
Junior and Student members.....		1,871.00	
Less: Cancellations.....		150.00	1,721.00
			\$27,990.20

## DUES—NEW MEMBERS

Members and Associates.....	3,439.50		
Less: 40% to Research Laboratory.....	1,375.80	2,063.70	
Junior and Student members.....		293.50	2,357.20

TOTAL DUES.....\$30,347.40

INITIATION FEES.....2,389.00

## INCOME FROM SALES OF EMBLEMS

AND CERTIFICATES FRAMES.....74.16

TOTAL INCOME FROM MEMBERS.....\$32,810.56

## FROM OTHER SOURCES

Editorial contract.....	\$13,333.30		
Less: Members' subscriptions.....	4,831.48	8,501.82	
Profit from sales of books, reprints, etc.....		443.03	
Sale of transactions.....		859.61	9,804.46

## FROM INVESTMENTS

Interest from savings accounts.....		82.92	
Interest from securities.....		158.15	
Interest from certificate of indebtedness.....		3.74	244.81

TOTAL INCOME.....\$42,859.83

## EXPENSES

## OFFICERS AND COUNCIL EXPENSES

President's Fund.....	\$ 1,742.36		
Council meetings, travel.....	2,062.12	\$ 3,804.48	

## COUNCIL COMMITTEES

Membership.....	366.45		
Meetings.....	1,247.49	1,613.94	

## SPECIAL COMMITTEES

Chapter Relations—Speakers' Bureau.....	\$ 177.18		
Chapter Relations—Delegates' travel.....	1,937.93		
Chapter Relations—Chapter records.....	186.34		
War service.....	671.17		
Chapters development.....	256.88	3,229.50	
		<u>\$ 8,647.92</u>	<u>\$42,859.83</u>

## DIRECT EXPENSES

A.S.A. membership.....	100.00		
Transactions.....	4,774.64		
Membership certificates.....	167.22		
Initiation fees to Reserve Fund.....	2,389.00		
Medals and awards.....	20.00	7,450.89	

## APPORTIONABLE EXPENSES

Salaries—Secretary and staff.....	16,516.50
Provision for additional compensation.....	3,471.52
Traveling—Secretary and staff.....	1,002.15
Rent and light.....	3,004.78
Telephone.....	576.04
Telegrams.....	99.67
Postage.....	1,563.49
General printing.....	694.77
Addressing and address changes.....	75.70
Office supplies.....	478.87
Professional services.....	600.00
Depreciation of furniture and fixtures.....	274.41
Bank charges.....	61.90
General office expenses.....	524.91

	28,944.71		
Less: 30% applicable to Guide.....	8,683.41	20,261.30	36,360.08

NET INCOME FROM SOCIETY ACTIVITIES..... \$ 6,499.75

## BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS  
New York, N. Y.

(For the Period from January 1, 1943, to October 31, 1943)

## INCOME

## MEMBERSHIP INCOME

## RENEWALS

	Actual	Budget Provision	Increases Decreases
100—Dues—Members.....	\$16,775.94	\$14,580.00	\$ 2,195.94
101—Dues—Associates.....	8,317.76	7,560.00	757.76
102—Dues—Juniors.....	1,676.00	2,000.00	324.00
103—Dues—Students.....	45.00	75.00	30.00

## NEW MEMBERS

104—Dues—Members.....	1,339.80	1,080.00	259.80
105—Dues—Associates.....	723.90	540.00	183.90
106—Dues—Juniors.....	230.00	200.00	30.00
107—Dues—Students.....	63.50	15.00	48.50

	2,357.20	1,835.00	522.20
TOTALS.....	<u>\$29,171.90</u>	<u>\$26,050.00</u>	<u>\$ 3,121.90</u>

## NEW MEMBERS

108—Initiation fees.....	2,389.00	1,700.00	689.00
109—Income—Emblems.....	14.42	20.00	5.58
110—Income—Certificate frames.....	59.74	25.00	34.74

TOTAL INCOME FROM MEMBERS.....	31,635.06	27,795.00	3,840.06
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## INCOME FROM PUBLICATIONS

115—Editorial contract.....	13,333.30	13,333.33	.03
116—THE GUIDE.....	1,469.30	70.00	1,539.30
117—TRANSACTIONS.....	859.61	500.00	359.61
118—Books and reprints.....	443.03	250.00	193.03
119—Codes.....	—0—	25.00	25.00

	16,105.24	14,038.33	2,066.91
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## INCOME FROM INVESTMENTS

125—Interest—Savings accounts.....	82.92	200.00	117.08
126—Interest—Securities.....	158.15	750.00	591.85
127—Interest—Certificate of indebtedness..	3.74	2.00	1.74

	244.81	952.00	707.19
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TOTAL CURRENT INCOME.....	\$47,985.11	\$42,785.33	\$ 5,199.78
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COLLECTION OF PRIOR YEAR'S DUES.....	1,175.50	1,500.00	324.50
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TOTAL INCOME.....	\$49,160.61	\$44,285.33	\$ 4,875.28
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## EXPENSES

## MEMBERSHIP

## OFFICERS' AND COUNCIL EXPENSE

150—President's Fund.....	\$ 1,742.36	\$ 1,500.00	\$ 242.36
151—Council meetings—Travel.....	2,062.12	2,500.00	437.88

	3,804.48	4,000.00	195.52
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## COUNCIL COMMITTEES

160—Executive.....	—0—	100.00	100.00
161—Finance.....	—0—	200.00	200.00
162—Membership.....	366.45	350.00	16.45
163—Meetings.....	1,247.49	1,200.00	47.49
164—Standards (includes codes).....	—0—	150.00	150.00

	1,613.94	2,000.00	386.06
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## SPECIAL COMMITTEES

170—Admission and advancement.....		150.00	150.00
171—Constitution and By-Laws.....		150.00	150.00
172—Nomination.....		150.00	150.00
173A—Chapter Relations—Speakers Bureau	177.18	1,000.00	822.82
173B—Chapter Relations—			
Delegates Travel.....	1,937.93	1,500.00	437.93
Chapter records.....	186.34	500.00	313.66
174—War service.....	671.17	1,500.00	828.83
Chapter development.....	256.88	—0—	256.88

	3,229.50	4,950.00	1,720.50
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## DIRECT EXPENSES

200—Membership subscriptions H.P.A.C. . .	4,831.48	4,583.33	248.15
201—A.S.A. membership.....	100.00	100.00	—0—
202—TRANSACTIONS.....	4,774.64	3,400.00	1,374.64
203—Membership roll (Yearbook).....	—0—	1,500.00	1,500.00
204—Membership certificates.....	167.22	200.00	32.78
205—Initiation fees to Reserve Fund.....	2,389.00	1,700.00	689.00
206—Medals and awards.....	20.00	100.00	80.00

	12,282.34	11,583.33	699.01
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## AFFORTIONABLE EXPENSES

210—Salaries—Secretary and staff.....	\$16,516.50	\$21,500.00	\$4,983.50
211—Provisions for additional compensation.....	3,471.52	3,000.00	471.52
212—Traveling—Secretary and staff.....	1,002.15	1,000.00	2.15
213—Rent and light.....	3,004.78	3,600.00	595.22
214—Telephone.....	576.04	600.00	23.96
215—Telegraph.....	99.67	250.00	150.33
216—Postage.....	1,563.49	1,500.00	63.49
217—General printing.....	694.77	800.00	105.23
218—Office supplies.....	478.87	500.00	21.13
219—Addressing and address changes.....	75.70	150.00	74.30
220—Professional services.....	600.00	700.00	100.00
221—Bank charges.....	61.90	50.00	11.90
222—Depreciation on furniture and fixtures.....	274.41	250.00	24.41
223—General office expense.....	524.91	500.00	24.91
CREDIT—30 % to GUIDE.....	8,683.41	10,320.00	1,636.59
	20,261.30	24,080.00	3,818.70
TOTAL EXPENSES.....	\$41,191.56	\$46,613.33	\$5,421.77

## STATEMENT OF INCOME AND EXPENSES—GUIDE

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS  
NEW YORK, N. Y.

(For the Period from January 1, 1943 to October 31, 1943)

## INCOME

Guide copy sales.....	\$22,672.48
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## EXPENSES

## FINAL COST OF 1943 GUIDE

Copy sales promotion.....	\$ 2,640.01
Indexing.....	411.92
Distribution.....	3,622.06
	\$ 6,673.99

AFFORTIONABLE EXPENSES.....	8,683.41	
CHAPTER MEETING ALLOWANCES.....	845.78	
SPECIAL APPROPRIATION TO RESEARCH.....	5,000.00	21,203.18
NET INCOME FROM GUIDE.....		\$ 1,469.30

## BUDGET COMPARISON—GUIDE

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS  
NEW YORK, N. Y.

(For the Period from January 1, 1943 to October 31, 1943)

	Actual	Budget Provision	Increases Decreases
CURRENT OPERATIONS			
INCOME			
301—Guide copy sales.....	\$22,672.48	\$20,500.00	\$ 2,172.48
EXPENSES			
FINAL COST OF 1943 GUIDE			
320—Copy sales promotion.....	\$ 2,640.01	\$ 2,500.00	\$ 140.01
321—Indexing.....	411.92	350.00	61.92
322—Distribution.....	3,622.06	1,500.00	2,122.06
TOTAL.....	\$ 6,673.99	\$ 4,350.00	\$ 2,323.99

**EXPENSES**

326—Apportionable expenses.....	8,683.41	10,220.00	1,636.59
327—Chapter meeting allowances.....	845.78	900.00	54.22
Special appropriation to Research....	5,000.00	5,000.00	—0—
<b>TOTAL EXPENSES.....</b>	<b>\$21,203.18</b>	<b>\$20,570.00</b>	<b>\$ 633.18</b>
<b>NET INCOME FROM GUIDE.....</b>	<b>\$ 1,469.30</b>	<b>\$ 70.00</b>	<b>\$ 1,539.30</b>

**DEFERRED OPERATIONS****INCOME**

300—Guide advertising.....	\$ —0—	\$23,000.00	\$23,000.00
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**EXPENSES****PRODUCTION AND DISTRIBUTION**

305—Paper.....	\$ 2,561.11	\$ 4,000.00	\$ 1,438.89
306—Printing and binding.....	—0—	14,000.00	14,000.00
307—Engraving and artwork.....	—0—	450.00	450.00
308—Editorial salaries.....	1,866.04	2,200.00	333.96
309—Distribution to members.....	—0—	2,000.00	2,000.00

**ADVERTISING SALES PROMOTION**

315—Salaries and commissions.....	2,750.00	3,600.00	850.00
316—Traveling.....	806.57	1,200.00	393.43

**EDITORIAL**

325—Guide committee expenses.....	332.36	350.00	17.64
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<b>TOTAL EXPENSES.....</b>	<b>\$ 8,316.08</b>	<b>\$27,800.00</b>	<b>\$19,483.92</b>
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L. T. Avery, Cleveland, Ohio, Chairman of the Constitution and By-Laws Committee, was called upon to present the proposed amendments to the Regulations Governing the Committee on Research.

# AMENDMENTS TO THE REGULATIONS GOVERNING THE COMMITTEE ON RESEARCH OF THE

## AMERICAN SOCIETY OF HEATING-AND VENTILATING ENGINEERS

These amendments are submitted in accordance with the terms of Article B-1, Section 10.

**ARTICLE I—Function**

*Section 1.* The function of the Committee on Research is the determination and dissemination of knowledge pertaining to the arts and sciences of heating, ventilating and air conditioning, and the equipment and apparatus utilized by the profession.

*Section 2.* The purpose of the Committee on Research are: (a) to collect, tabulate, co-ordinate and determine by laboratory research, data pertaining to the arts and sciences of heating, ventilating and air conditioning; (b) to establish and maintain a Research Laboratory, and to make arrangements for cooperative research work with universities, colleges and other appropriate organizations.

**ARTICLE II—Organization**

*Section 4. Research Finance Committee*—The Chairman of the Committee on Research shall appoint for the ensuing year a Research Finance Committee of five (5) members of the Society and shall designate the Chairman. The Director of Research shall be an ex-officio member of this Committee.

*Section 5. Director of Research*—There shall be a Director of Research appointed by the Committee on Research subject to approval by the Council.

*Section 7. Technical Advisory Committees*—The Chairman of the Committee on Research shall appoint such Technical Advisory Committees and designate a Chairman of each, as may be deemed

advisable, to act in an advisory capacity to the Committee on Research and the Director of Research for specific projects under consideration. At least one (1) member of each Technical Advisory Committee shall be a member of the Committee on Research. In addition the Director of Research and the Chairman of the Committee on Research shall be ex-officio members of all Technical Advisory Committees.

#### ARTICLE III—Duties of Officers

*Section 1. Chairman*—The Chairman of the Committee on Research shall preside at all meetings of the Committee and of the Research Executive Committee. He shall be an ex-officio member of the Council. He shall approve and sign all contracts made on behalf of the Committee on Research which, under the Constitution and By-Laws of the Society, require the signatures of the President and Secretary of the Society before becoming operative.

*Section 2. Vice-Chairman*—In the event that the Chairman of the Committee on Research is unavailable, the Vice-Chairman shall possess all the powers and perform all the duties of the Chairman.

*Section 3. Director of Research*—The Director of Research shall be responsible to the Committee on Research. He shall supervise all laboratory activities of the Society and shall have general charge of all research activities including the making of contracts for the rental or purchase of equipment or materials. He shall, under the authority of the Committee, select and engage, when necessary, a supervisor of the Research Laboratory and such research assistants and other personnel as may be required.

The Director of Research shall, subject to the approval of the Research Executive Committee, determine the order in which the subjects shall be investigated by the Research Laboratory.

The Director of Research shall prepare for the Research Executive Committee a budget of estimated income and expenditures of the Committee on Research for the next fiscal year.

All bills against Research which are covered by the budget shall, before being presented to the Secretary of the Society for payment, be approved by the Director of Research.

*Section 4. Technical Adviser*—The Technical Adviser shall be the consultant and adviser to the Committee on Research and the Director of Research on such matters as may be submitted to him by the Chairman of the Committee or the Director.

#### ARTICLE IV—Duties of Committees

*Section 1. Committee on Research*—The Committee shall establish and maintain a research laboratory, at a location and under conditions to be approved by the Council of the Society.

The Committee shall, subject to the approval of the Council, select and engage a Director of Research, and fix his compensation. The Committee shall establish a research program; shall approve a budget which shall govern expenditures for the fiscal year and shall approve all cooperative agreements.

##### *Section 2. Research Executive Committee.*

(b) The Research Executive Committee shall before November first of each year adopt a budget of estimated income and expenditures of the Committee on Research for the next fiscal year. Any proposed expenditure of Research funds outside of the approved budget shall be approved by the Executive Committee or, by delegation, by the Chairman of the Committee on Research, before the expenditure is made.

(c) The Research Executive Committee shall approve the compensation of all employees of the Committee on Research.

*Section 3. Research Finance Committee*—The Research Finance Committee shall with the cooperation of the Director of Research solicit the contribution of funds for the maintenance of Research activities. The Chairman shall report on the progress of the work at frequent intervals to the Chairman of the Committee on Research.

*Section 4. Technical Advisory Committees*—Technical Advisory Committees shall act in an advisory capacity to the Committee on Research and the Director of Research on all subjects referred to them. All Technical Advisory Committees shall be governed by the "Rules for Technical Advisory Committees" (Appendix A1).

#### ARTICLE V—Government

*Section 5. Cooperative Agreements*—At the discretion of the Committee on Research cooperative agreements may be entered into with universities, colleges or other appropriate organizations, for the investigation of any specific subjects on the research program. When such a cooperative agreement is to be consummated the Research Executive Committee shall prepare a contract, stating the terms agreed upon. This contract shall be signed by the authorized representative of the college, university or organization, by the Chairman of the Committee on Research on behalf of the Committee, and by

the President and Secretary of the Society, as required by Article B-VI, Sections 1 and 5 of the By-Laws of the Society.

*Section 6. Headings of Papers*—All papers, findings or reports resulting from the work of the Committee on Research shall, when published by the Society, be headed as follows:

(a) No change.

(b) No change.

(c) If the paper, finding, or report is the result of work at the Society's Research Laboratory, when located elsewhere than in (a) the following statement shall be used: "This paper is the result of research carried out by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at (address)."

*Section 7. Payment of Bills*—(a) All bills against Research activities shall be approved by the Director of Research, and he shall present approved bills to the Secretary of the Society. The Secretary shall check and, if correct, pay all items when covered by the budget, or emergency expenditures approved in writing by the Chairman of the Committee on Research. A summary of these accounts shall be made at least every month on a form approved by the Council. The Treasurer of the Society shall issue a check against the Research Fund, payable to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, which check shall be deposited in the Secretary's special account. The Secretary shall draw against this amount in settlement of all approved expenditures.

#### ARTICLE VII—Technical Advisory Committees

*Section 1.* The Rules for Technical Advisory Committees may be amended at any regular meeting of the Committee on Research by a two-thirds vote of the members present, provided that any such amendment shall have been submitted in writing to each member of the Committee on Research at least two weeks before the meeting at which action is to be taken. Any such amendment shall become effective immediately upon its adoption by the Committee on Research and shall be published in the next issue of the JOURNAL.

## Appendix A1

### Rules for Technical Advisory Committees <sup>†</sup>

#### I. Organization of Committee

(1) *Creation*: The Chairman of the Committee on Research shall appoint such Technical Advisory Committees and designate a Chairman of each, as may be deemed advisable, to act in an advisory capacity to the Committee on Research and to the Director of Research for specific projects which are authorized.

(2) *Scope*: The Chairman of the Committee on Research shall furnish a brief statement covering the scope of activity for each Technical Advisory Committee at the time of appointment.

(3) *Membership*: The Chairman of each Technical Advisory Committee shall be a member of the Society and at least one member of the committee shall be a member of the Committee on Research. The Chairman of the Committee on Research and the Director of Research shall be ex-officio members of all Technical Advisory Committees.

Additional members may be appointed to an Advisory Committee at any time by the Chairman of the Committee on Research. The committee personnel shall be selected so that a majority are members of the Society.

The term of office of members of all appointed committees shall end at the close of each Annual Meeting and any member shall be eligible for re-appointment.

(4) *Sub-Committees*: The Advisory Committee shall decide what sub-committees shall be established and shall define the duties of each. Sub-committees shall work only upon such problems as are assigned to them or are authorized by the Advisory Committee.

#### II. Officers and Meetings

(1) *Officers*: The Chairman of each Technical Advisory Committee may appoint a Vice-Chairman and Secretary.

The Chairman, or in his absence the Vice-Chairman or designated member, shall preside at all meetings of the committee and shall be an ex-officio member of all sub-committees.

Minutes shall be kept of all committee meetings and copies shall be sent to the Chairman of the Committee on Research, the Director of Research and the Secretary of the Society.

<sup>†</sup> Adopted by Committee on Research, January 30, 1944.

(2) *Meetings:* Regular meetings of the committee should be held during the Annual and Semi-Annual meetings of the Society or special meetings may be called by the Chairman. The meetings of any Technical Advisory Committee should be open only to the members of that Committee and to such visitors as may be approved by the Chairman.

A quorum for the transaction of business shall consist of a majority of the committee membership.

The date and place of all committee meetings shall be determined by the Chairman and written notices of all meetings shall be mailed to the members of the committee at least two weeks in advance of the meeting.

The Chairman of the Advisory Committee shall be notified in advance of all meetings of sub-committees and shall receive copies of minutes of these meetings.

### III. Cooperation with Other Organizations

(1) *Method of Initiating Cooperation:* An Advisory Committee desiring to cooperate with representatives of other organizations or desiring to invite the cooperation of other organizations in the work being done, shall address a recommendation to that effect to the Research Executive Committee and, if approval is given, negotiations to the desired end shall be conducted by the Chairman of the Committee on Research.

### IV. Expenses

(1) *Current Expenses:* Expenses for postage incurred in connection with the business of committees shall be refunded by the Society on presentation of vouchers approved by the Chairman.

(2) *Stationery:* Correspondence relating to the business of committees shall be conducted on official stationery of the Society which will be furnished upon request.

(3) *Extraordinary Expenses:* Items of expense other than postage will not be assumed by the Society, unless such expenditures were incurred in pursuance of previous authorization by the Committee on Research and within the limits specifically fixed by the Committee on Research.

(4) *Special Funds:* Advisory Committees may be authorized by the Research Executive Committee to solicit contributions to be earmarked for research on specific projects approved by the Committee on Research. All funds thus collected shall be transmitted to the Secretary of the Society and deposited in the Research Fund subject to disbursement only on voucher approved by the Director of Research.

(5) *Salaries and Fees:* Committees are not authorized to pay salaries or professional fees in any form to any of their officers or members.

### V. Directions for Conduct of Work

(1) The Committee shall first review all previous work done on the subject assigned to them for study and prepare a bibliography.

(2) It shall outline a comprehensive program of research and indicate fields in which most productive work should be done.

(3) It shall select research most suitable for Advisory Committee activity and decide what phases of the program require laboratory investigation.

(4) It may consult the Technical Advisor to the Committee on Research for advice concerning any phase of its projected or approved research program.

(5) The Committee shall survey available laboratory facilities and recommend to the chairman of the Committee on Research those places which are best equipped with apparatus and personnel to undertake the desired research.

(6) The Committee shall, with the cooperation of the Director of Research, prepare a statement of the scope of the investigation and this, together with an outline of the method of procedure to be followed and an estimate of the cost of the work, shall be transmitted to the Chairman of the Committee on Research for approval.

(7) When a project has been approved, the Chairman of the Technical Advisory Committee shall share, with the Chairman of the Committee on Research and the Director of Research, the responsibility for supervision of the work.

(8) Laboratory investigations may be conducted: (a) At the Society's Laboratory with general funds appropriated by the Committee on Research or earmarked funds specifically collected for particular projects. (b) Through a Cooperative Agreement with an institution approved by the Committee on Research with general funds appropriated by the Committee on Research or earmarked funds specifically collected for particular projects.

(9) All sponsored or supervised research planned by an Advisory Committee must be designed to develop fundamental principles and/or test methods and, in the publication of the results by the Society or with the approval of the Society, no specific commercial product shall be identified.

(10) An Advisory Committee is not empowered to draft Society codes but it may draft regulations for a testing procedure for Committee use.

(11) During the progress of any investigation the Advisory Committee Chairman shall assist the Chairman of the Committee on Research in disseminating through the headquarters office of the Society, proper publicity regarding the projects.

(12) Each Advisory Committee shall review each year those sections of **THE GUIDE** which pertain to its scope of activity and submit its recommendations in writing to the Guide Publication Committee.

(13) Each Advisory Committee shall submit an annual report in writing to the Chairman of the Committee on Research, not later than December first of each year, covering the work of the committee during the year with recommendations for future activity.

#### *VI. Submittal of Research Papers*

(1) When Laboratory work has been completed and a report prepared covering the work or any other report is completed covering a phase of the Committee's activities, a copy of the manuscript, together with a ballot blank, shall be sent to each member of the Advisory Committee not less than 60 days prior to publication or meeting presentation date, whichever is the earlier.

(2) Committee members shall be requested to review the paper or report and return a ballot to the Chairman within 10 days.

(3) Within a period of 10 days the Chairman of the Advisory Committee may obtain necessary revisions to the manuscript based on the comments received from Committee members and resubmit the manuscript for final Committee approval.

(4) If two-thirds of the voting Committee members approve the paper it shall be recommended to the Director of Research and Chairman of the Committee on Research for publication.

(5) Manuscripts approved for publication by the Chairman of the Committee on Research shall be transmitted to the Secretary of the Society not less than 30 days prior to publication and submitted in the form outlined in **SUGGESTIONS FOR AUTHORS** adopted by the Society.

(6) Research papers are subject to final approval for publication as per Article R-XII of the **RULES** of the Society.

#### *VII. Reproduction of Research Data*

*General:* It is the intention of the Committee on Research that all reports covering investigations at the Research Laboratory and in cooperative institutions receive the widest dissemination possible. However, in the interests of everyone concerned it is important that these research data be used in proper form so that the facts may not be distorted. It is for this reason that these rules have been prepared covering the reproduction of Society research data.

##### *Publications*

(1) Any publication wishing to reprint a research report in part or in full must first obtain permission from the Society in writing. After permission is obtained, the material to be reproduced must be submitted for final approval to the Secretary of the Society at least 10 days in advance of publication and must contain the authorized credit reference to the authors and the Society.

##### *Reprints*

(2) Reprints of any research report may be obtained by application to the Society within a reasonable time after publication and upon payment of charges established by the Council, for distribution in such form by any individual, company or trade association.

##### *Handbooks, Textbooks or Company Manuals or Promotional Trade Literature*

(3) Data, charts, or quoted text, including discussion of same, from any research report; or any data, charts, or quoted text included in **THE GUIDE** which has been prepared from any research report; may be reprinted in any handbook, text book or company or trade association engineering manual or promotional trade literature with the authorized credit reference to the original source of publication, after permission for such reproduction is obtained from the Society in writing.

##### *Advertisement*

(4) The use of any data, charts, or quoted text from any research report or **THE GUIDE** for display advertising purposes in any magazine or other paid advertisement cannot be permitted.

W. L. Fleisher, New York, N. Y., in discussing the proposed amendments stated that in order to keep a contact between the Committee on Research and the Council, the Society adopted an amendment several years ago making the Chairman of the Committee on Research *Ex-Officio*, a member of the Council. He mentioned that whereas it had been unheard of that the Committee on Research and the Council should be interlocking, there were now 6 Council members on the Committee on Research. The Society should have enough capable members, he said, to fill the positions on both the Committee on Research and the Council without need for duplication. In his opinion

it was the intent of the Society that the only representative on the Council from the Committee on Research should be its chairman. In order to maintain independence of the Committee on Research he suggested an amendment to the By-Laws somewhat as follows:

No member shall be eligible to run for two elective offices at any one election.

President Blankin stated that the change proposed by Mr. Fleisher would necessitate a change in the By-Laws.

Mr. Avery said that the change proposed by Mr. Fleisher would prevent a member from running for office while holding another office. Careful consideration should, therefore, be given to all possible effects of the proposed change. It seemed possible, however, to obtain an informal opinion from the members present at this meeting.

Mr. Fleisher suggested that with such a large attendance it would be a good time to bring the matter to the Society's attention and he stated that an expression from those who had served on both the Council and the Committee on Research would be desirable.

President Blankin stated that, as Mr. Avery had indicated, the amendments to the Regulations Governing the Committee on Research were recommended for adoption by unanimous action of the Committee on Research.

On motion by L. T. Avery, seconded by W. A. Russell, it was

VOTED that the amendments to the Regulations Governing the Committee on Research be approved.

On motion by W. L. Fleisher, seconded by F. W. Legler, Minneapolis, it was

VOTED that the Committee on Constitution and By-Laws consider necessary changes in the Constitution and By-Laws and Rules of the Society which would prevent duplication of members on the Council and the Committee on Research.

President Blankin turned the meeting over to First Vice-Pres. S. H. Downs, Kalamazoo, Mich., who called upon John Howatt, Chicago, Ill., to read the paper, Fifty Years in Heating and Ventilating which was prepared by S. R. Lewis (see p. 81).

Vice-President Downs then presented Lieut.-Comdr. F. C. Houghten, who gave an abstract of his paper on Progress in the Development of Standards for Comfort Air Conditioning (see p. 87).

#### PANEL DISCUSSION

Vice-President Downs called upon Dr. B. M. Woods, Berkeley, Calif., to conduct the Panel Discussion on the subject of Future Trends of Heating, Ventilating and Air Conditioning.

Chairman Woods called the roll of panel members as follows: D. M. DeBard, War Production Board, Washington, D. C.; W. H. Driscoll, Syracuse, N. Y.; L. T. Mart, Kansas City, Mo.; Comdr. T. H. Urdahl, Washington, D. C.; and C. E. Lewis, Rochester, N. Y.

Chairman Woods presented the plan which was to have a member of the Panel introduce each of the subjects, ventilation, heating, cooling and physiological effects. He stated that a fifth item, war conditions and conservation would be discussed separately.

Chairman Woods expressed his thanks to the members of the Panel and to those who aided in the discussion.

### THIRD SESSION—TUESDAY, FEBRUARY 1, 9:30 A.M.

The third session was called to order at 9:30 a.m. by President Blankin in the Georgian Room, and Second Vice-Pres. C.-E. A. Winslow, New Haven, Conn., then took the chair.

F. E. Giesecke, College Station, Tex., presented an abstract of his paper on A Study of Intermittent Heating of Churches (see p. 99).

Chairman Winslow then called for the second paper on The Resistance to Heat Flow Through Finned Tubing, by W. H. Carrier and S. W. Anderson (see p. 117).

Chairman Winslow introduced Prof. G. L. Tuve who presented an abstract of the paper on Control of Air-Streams in Large Spaces (see p. 153), which he prepared with G. B. Priester.

Chairman Winslow made the closing remark that many of the difficulties mentioned in the papers presented by Dr. Giesecke and Professor Tuve would be non-existent if buildings were heated by radiant heat rather than by convection heating. He then thanked the various authors and those who took part in the discussions, whereupon the meeting adjourned at 12:30 p.m.

### FOURTH SESSION—TUESDAY, FEBRUARY 1, 2:00 P.M.

Vice-Pres. S. H. Downs called the fourth session to order at 2:00 p.m. and presented Prof. F. B. Rowley, who gave an abstract of his paper on Discoloration Methods of Rating Air Filters (see p. 173), which he prepared with his associate R. C. Jordan.

Vice-President Downs then called for the next paper on the subject of The Axial Flow Fan and Its Place in Ventilation which was prepared by W. R. Heath and A. E. Criqui, and which was presented in abstract by Mr. Heath (see p. 197).

Vice-President Downs then presented T. H. Troller, who gave an abstract on his paper on The Aerodynamic Development of Axial Flow Fans (see p. 213).

### FIFTH SESSION—WEDNESDAY, FEBRUARY 2, 9:30 A.M.

The fifth session was called to order by President Blankin at 9:30 a.m. in the Georgian Room. He presented Major Arthur Nelson of the Corps of Engineers, U. S. Army, who spoke on the subject of Fuel Economy and Army Heating.\*

President Blankin then introduced George T. Seabury, Secretary, *American Society of Civil Engineers*, who spoke on the subject of Collective Bargaining for Engineering Employees.<sup>†</sup> Mr. Seabury reviewed the history of the approach of collective bargaining as observed by the *ASCE*. Attention of the *ASCE* was first directed to the inroads of labor organizations in 1936, and consequently, a committee was formed in 1937 to determine the extent of such

\* Address published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, April 1944, p. 232.

† Published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, April 1944, p. 236.

inroads. The committee found that a number of pressure groups, some related to and independent of organized labor were active and wrote a report which was published by the *ASCE* in 1938.

Among the significant statements appearing in the report were the following: "Membership in a trade union is primarily an economic matter, therefore, the *ASCE* should consider such membership as having no more bearing upon a man's qualifications for membership in the Society than for his religious or political affiliations. The Wagner Labor Relations Act has encouraged the extension of existing unions, formation of others, and has paved the way for complete unionization of the employees of many industries and it may be expected to result in the formation of collective bargaining groups in all plants and offices where any considerable number of people are employed. Engineers and architects, as well as draftsmen and other professional men are not and cannot be exempt from the provisions of the Act."

As a result of investigation made throughout the United States, the Board of Direction of the *ASCE* requested the committee on employment conditions to establish a program which included instructions for local sections to make provisions for collective bargaining groups to be composed of and controlled by *ASCE* members, but including non-members who would be eligible to join the group. At the conclusion of his address Mr. Seabury answered a variety of questions which were asked by members of the audience.

#### SIXTH SESSION—WEDNESDAY, FEBRUARY 2, 2:00 P.M.

President Blankin called the meeting to order and stated that the paper to be presented by Prof. F. W. Hutchison, Berkeley, Calif., was a result of cooperative research between the Society and the University of California. Professor Hutchinson presented an abstract of the paper on Optimum Surface Distribution in Panel Heating and Cooling Systems (see p. 231).

President Blankin stated that permission had been asked for a discussion from the floor concerning the Kilgore Bill on which hearings were now being held in Congress.

W. C. RANDALL, Detroit, Mich.: Some time ago I received a copy of an abstract of the Kilgore Bill from the *National Association of Manufacturers*. I brought this to the attention of the Board of Governors of Michigan Chapter which passed a resolution expressing the opinion that Bill SO2 and its companion measure HR2100 now pending are undesirable and requested that the Society give consideration to this matter at the 50th Annual Meeting.

The Michigan Chapter was interested in determining the reaction of the national organization of A.S.H.V.E. toward the bill.

PRESIDENT BLANKIN: This was first brought to the Society's attention by the Delta Chapter and was discussed by the Council and referred to the War Service Committee. I will ask Mr. Howatt to make a report.

JOHN HOWATT, Chicago, Ill.: The Committee has kept itself informed on the progress of the Bill and as there appeared to be some possibility that lately it had been given added impetus, it should be referred to the new A.S.H.V.E. Council for consideration of the various features which it contained. Senator Kilgore promised to communicate with me regarding the date for a hearing on the Bill. I feel that the Society should not adopt a resolution expressing an opinion of the Bill at this time, but that the matter should be referred to a Committee for study.

L. P. SAUNDERS, Lockport, N. Y.: It is my opinion that very little attention would be paid to any resolution which might be passed and consequently the only

effective way to influence legislation would be for each member to write directly to his congressman.

FERDINAND JEHL, Indianapolis, Ind.: *The American Association for the Advancement of Science* is against the Bill.

W. A. DANIELSON, Memphis, Tenn.: The effect which centralized control of learning is disastrous and I feel that such control should not be permitted to the slightest degree.

R. A. MILLER, Pittsburgh, Pa.: I stress the importance of taking action regarding the Bill.

On motion of Mr. Howatt, seconded by W. L. Fleisher, it was voted:

THAT the President appoint a Committee to study the Kilgore Bill and to take appropriate action regarding it.

President Blankin recognized R. A. Folsom, San Francisco, chairman of the Resolutions Committee, who presented the following resolutions which were unanimously approved.

### Resolutions

*Whereas*, our President, Merrill F. Blankin, during the past year has devoted his untiring efforts to the advancement and well being of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and

*Whereas*, the results of his able and continuous concentration on Society duties at very great sacrifice to his other business and personal affairs, have been manifest in the virility and the substantial progress of every phase of Society activities and,

*Whereas*, his visits to the various local chapters have been a source of inspiration and stimulation and have added greatly to the welding of the membership into a unified and closely knit organization,

*Be It Resolved*, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS expresses its thanks and appreciation for his capable leadership and also for the gracious assistance of Mrs. Blankin who accompanied him on many of his visits, and

*Be It Further Resolved*, that the thanks and appreciation of the Society also be conveyed to all of the other Officers and to the Council who have so diligently served its interests throughout the year.



*Whereas*, the administration of the Society's activities under the leadership of its officers has been so ably carried on by the Council, the Council Committees and the Advisory Council, and

*Whereas*, the special activities of the Society have been handled by Special Committees, the Committee on Research and the Technical Advisory Committees, and

*Whereas*, the Council, the Chairman, and the Members of these groups and Committees have at great personal sacrifice unstintingly given of their time in this work, and

*Whereas*, the members of the Society anticipate with pleasure the receipt of the new GUIDE on which the Publication Committee has worked so long and diligently,

*Now Be It Resolved*, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and its members assembled at this 50th Annual Meeting express their thanks and gratitude to those individuals who have so willingly served and who have been in such a large measure responsible for the continued advancement of the Society and its activities.

*Be It Resolved*, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS recognizes with the greatest of pleasure the presence at this 50th Anniversary Meeting of 19 of the 23 living past presidents and expresses its grateful appreciation for their continued interest and participation in its work, and

*Be It Further Resolved*, that an expression of good wishes and greetings be conveyed to our two Charter members, Homer Addams and Charles F. Hauss, together with the regrets of the membership that they were unable to attend this meeting.



*Whereas*, the 50th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS ending a half century of continued progress in the heating and ventilating engineering field is now about to be concluded, and

*Whereas*, the Technical Sessions, the Entertainment, the Committee Meetings and business matters presented, have been of unusual interest and exceptionally stimulating to those in attendance,

*Be It Resolved*, that an expression of thanks and appreciation be adopted and that copies of this resolution be published and sent to each of the persons and agencies who have contributed toward making this 50th Annual Meeting so successful and enjoyable.

To the Honorable Fiorello H. La Guardia, Mayor of the City of New York, for his able address on The Post War Era Offers a Challenge to Engineering;

To R. H. Carpenter for his services as toastmaster at the Get-together Luncheon;

To the Engineering Societies, Associations and individuals who sent representatives and messages of greetings and felicitations to the A.S.H.V.E. on the occasion of its 50th Anniversary;

To the Officers and Members of the New York Chapter and their ladies who have welcomed us and who through their cordial hospitality have made our visit so enjoyable;

To the Honorary Chairmen, Homer Addams, W. H. Carrier, W. H. Driscoll, W. L. Fleisher and D. D. Kimball;

To the General Chairman, Alfred J. Offner, and the Ladies' Co-Chairmen, Mr. and Mrs. H. J. Ryan, who have been most considerate of our every desire;

To the Chairmen and Members of all other Committees of the New York Chapter who have given so generously of their time to insure the success of this meeting;

To the Presiding Officers of the Technical Sessions and to Dr. Baldwin M. Woods, who guided the panel discussion;

To the authors and their associates who have prepared and presented the excellent technical papers;

To Mr. Samuel G. Hibben, Director of Applied Lighting of Westinghouse Electric and Manufacturing Co. for his entertaining demonstration and address on Tomorrow's Uses of Radiant Energy;

To the Management and Staff of the Hotel Pennsylvania for their fine cooperation and service and the special courtesy of arranging for the ladies to inspect the Statler Research Kitchen;

To the New York Convention and Visitors Bureau for their generous assistance and cooperation and the special service rendered in handling registration and furnishing special information about the City of New York;

To the New York newspapers for their coverage of this meeting, and to the trade magazines for their generous cooperation and the attendance of their representatives;

To N. B. C., Columbia Broadcasting Co., and Philco Corp., for their courtesy in furnishing tickets for several broadcasts, and a special greeting to Fred Waring and his company for their salute to the Society;

To Rockefeller Center for the opportunity afforded our members to inspect the mechanical equipment.

Respectfully submitted,

RESOLUTIONS COMMITTEE,

R. A. FOLSOM, *Chairman*

WILLIAM GLASS

H. BERKLEY HEDGES

President Blankin introduced Past Pres. W. T. Jones, Boston, who conducted the installation of the Society officers and new members of the Council for 1944. President-Elect S. H. Downs, Kalamazoo, Mich., was presented with the gavel and closed the meeting.

The meeting adjourned at 4:25 p.m.

#### GET-TOGETHER LUNCHEON

The Get-Together Luncheon on Monday was opened by Toastmaster R. H. Carpenter who introduced the Hon. F. H. LaGuardia, Mayor of the City of New York, who spoke on Post-War Era Offers a Challenge to Engineering.<sup>10</sup> The Mayor outlined the city's post war plans which will provide employment for 200,000 and he emphasized the fact that New York is further advanced on its post war planning program than any other City, State or the Federal Government. The Mayor indicated that the program would include schools, hospitals, courts, police and fire stations, sewage disposal facilities, playgrounds and street and highway improvements. Mayor LaGuardia told his audience that the city wanted to have the last word in heating and ventilating,

<sup>10</sup> Published in A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, April 1944, p. 240.

cleaning and sanitary appliances, and he also indicated that it was up to the engineers who have been creating things to insure the success of the public works program after, as it can't be done by amateurs, nor by lawyers or professional politicians.

#### PAST-PRESIDENTS' DINNER

At 6:30 Monday evening 16 Past Presidents gathered for dinner in Parlor C of the Hotel Pennsylvania and enjoyed a reunion that has become traditional.

#### MODERN USES OF RADIANT ENERGY

The control of mold and decay, the elimination of many of the hazards associated with darkness, even the extension of the human life span itself, were some of the benefits to be expected from new uses of radiant energy, was the prediction of Samuel G. Hibben, in a address Monday evening to a group of 500 members and ladies. Mr. Hibben, director of applied lighting at the Westinghouse Lamp Division, Bloomfield, N. J., said that ultraviolet radiations, such as those produced by an ultraviolet lamp, can reduce food spoilage by preventing the formation of mold and decay. These same radiations can also purify the air we breathe and the articles we touch with our lips, thus lessening the spread of many diseases and making the air we breathe indoors more healthful.

Engineers responsible for the creation of healthful indoor weather conditions may depend more and more upon radiant heat as a necessary adjunct in heating our homes and offices. This same radiant energy might also be used in the field of quick cooking and in the drying and preserving of foods. Radiant heat, because it can be quickly applied and easily localized, offers a saving in fuel and a desirable increase in human comfort.

#### OLD TIMERS REUNION

Nearly 500 members and ladies enjoyed the Old Timers Reunion on the Roof of the Hotel Pennsylvania at 9:00 p.m. Monday evening. As the members assembled, A. C. Buensod's Special Events Committee, with the assistance of a "special policeman" of 19th century vintage, escorted the men to a special parlor where their appearance was "improved" by the proper application of mustache and beard of the gay 90's. A buffet supper was served, and the group enjoyed square dancing and the more modern versions of the art.

#### BANQUET

The grand ballroom of Hotel Pennsylvania was the scene of a brilliant gathering of nearly 800 members and guests on the evening of Feb. 2. After the singing of the national anthem a group picture was taken and following the service of dinner, W. H. Driscoll, Syracuse, N. Y., toastmaster and past president of the Society, greeted those present and said that the dinner was a fitting climax to a fine meeting celebrating the 50th Anniversary of the Society.

TOASTMASTER DRISCOLL: We all know that the surest way for the Toastmaster to win your immediate applause is to announce the fact that there will be no speeches this evening. I am sorry that I cannot mislead you by any such erroneous statement. There are going to be some real speeches.

The Committee has given me the privilege of taking a little more time than usual because of the special occasion and because of my enthusiasm for the Society and for the people in the Society.

The first 50 years are the hardest! Our Society has come gloriously through that first half-century and stands tonight strong, vigorous, and well prepared for the service and career that lie before it.

Tonight on the occasion of its Golden Anniversary we gather here to celebrate the triumphs of the past and challenge the problems of the future.

We glory tonight in the memories of that handful of foresighted pioneers who, fifty years ago, determined that the time had come to raise the *trade* of steamfitting to the dignity of a *profession*, and to endow the uncertain art of heating and ventilating with the truth and certainty of a science.

We honor tonight the many fine men, past and present, who, interpreting the spirit of the Founding Fathers, have carried the Society to heights undreamed of by them—men who, through research and investigation, have unlocked the secrets of nature and revealed its hidden mysteries—men who have enriched the literature of the art in the many fine technical papers which they have presented before the Society, and men who, out of the wealth of their experience, have contributed to discussions at the Society's meetings and have helped in the administration of its affairs. Their name is legion, and tonight we salute them.

And to the young men who are heirs to all that has gone before, who have come to us with the spirit and courage of youth and with an advantage in modern education that starts them in where we leave off—these we offer as a challenge to the future, supreme in the confidence that they will not fail us but will carry the art, the science, and the Society to still more glorious successes in the future.

Far be it from me to cast a shadow on that brilliant future which we and they look forward to so hopefully, but I feel that we would be false to our trust if, out of the wisdom of age and experience, we failed to warn them that the path ahead is not easy and the problems not simple. Progress and achievement grow only out of work, and the will to work, the courage to tackle adversity and the spirit to rise above disappointments and discouragements.

The world has reached no such level of economic stability and security as will permit the slightest departure from sound business practice. We are a non-profit-making organization in which the Council acts as trustees of the funds of the Society. As such, its members have a moral obligation to be more scrupulous and careful with Society funds than they need be with their own. They are not warranted in mortgaging the future of the Society, however hopeful it may appear, by committing themselves to obligations and expenditures beyond their current funds and income. Our Society is subject to the same economic cycle that all individuals, organizations, and nations experience. It has its periods of prosperity and its periods of depression and, regardless of its successes, it should acquire no delusions of grandeur, but proceed humbly to fulfil its destiny within the limits of reason and common sense.

This is neither the time nor the place to present a detailed history of the Society nor a list of the men who have distinguished themselves in its work, but I am going to presume on your patience in an attempt to give you some understanding of the times and the circumstances that gave birth to the Society.

In 1889 the *National Association of Master Steamfitters*,—now known as the *Heating, Piping, and Air Conditioning Contractors' National Association*—was organized for the purpose of improving conditions in the industry, both in its commercial and technical aspects.

For several years thereafter the program of its national convention included technical articles by members and others. The business aspects of the industry, however, presented many problems that, in the minds of many members, outweighed in importance any matters of technical interest.

In June, 1894, when D. M. Nesbitt came from London to deliver a prepared technical paper, so little time was devoted to discussion of the subject that Hugh J. Barron, a rough and rugged member of the association, and who, I feel, must be recognized as the real father of this Society, wrote a vitriolic letter to the editor of *Heating and Ventilation*. Soon afterward a conference between Messrs. Barron, L. H. Hart (the editor of *Heating and Ventilation*) and W. M. Mackay, was

arranged and the object of this meeting was to consider ways and means of divorcing the technical side of the industry from the commercial side, so that each could pursue its own way free and untrammelled.

Within three weeks these men had worked up enough interest to bring together sixteen men at a meeting held in Mr. Hart's office in the Pulitzer Building, one of New York's so-called skyscrapers, now almost lost in the canyons of the lower city.

This meeting was held on August 2, 1894, and before it adjourned all present signed an agreement to become members of a heating and ventilating engineers' society. The idea of having two organizations, one devoted to business functions and application problems of the growing industry, and the other concerned with the engineering plans was hailed as a happy solution.

A committee of five was appointed to formulate plans for organizing such a society, and they set to work with such a will that, by August 20, their plans were formulated and September 10, 1894, was selected as the date of the first general meeting.

This meeting was held in the Broadway Central Hotel, and, of the 150 or more to whom invitations were sent, 75 responded and signed up as charter members. Before the meeting was over, a constitution was adopted, the present name of the Society was decided upon, and plans were made for the first annual meeting. It is significant that the first Constitution contained a provision that contractors were not eligible as members of the new Society and this remained in force for some time, so that there would be no change in the status quo of the contractors' organization.

E. P. Bates, Syracuse, N. Y., was elected temporary president to serve until the end of the first annual meeting. This meeting was held on January 22, 23, and 24, 1895, in the hall of the American Society of Mechanical Engineers, then located at 12 West 31st Street, New York City.

That was the first annual meeting, and Stewart A. Jellett, of Philadelphia, was elected the first permanent president. It seems a very happy coincidence that the president serving at this Fiftieth Annual Meeting, Mr. Merrill A. Blankin, is also a worthy citizen of the Quaker City.

I well remember Hugh Barron, who was exceedingly active in Society affairs up until his death, and if you go back through the Transactions of the early years of the Society, you will find his name appearing many times in discussions of technical papers and in matters of parliamentary procedure.

His favorite remark, which I heard many times, was, "Mr. Chairman, I object." He was a great objector, but in a constructive way; and, although gruff in manner, a man of high ideals who injected into the Society much of the spirit it has since retained.

Now let me bring up at this moment some memories of Little Old New York as it was in the days when the stork delivered our baby Society into its sizzling atmosphere. It was the late summer of 1894; nevertheless, if you are to believe the stories that will be related in this very hotel on the 12th of next month by the Blizzard Men of '88, I am sure you will be convinced that New York was still digging its way out of the greatest snowstorm that ever occurred in the history of the human race, one that descended on the town six years earlier.

At any rate, the nation was still licking the wounds it had suffered from what was then called the Panic of '93. How little did those old-timers know of the depths to which a depression may go! Grover Cleveland was still presiding over the destinies of the nation; but as a result of the depression his Democratic party was tossed out of the White House at the next election. Man, however, is a fickle individual, and the effect of our last depression was that he tossed the same party right back in again—and, God help us! it is with us yet.

Thomas Gilroy was the Mayor of Manhattan Island, because Manhattan Island constituted all there was of the City of New York. Greater New York did not come into being until three or four years later. Brooklyn was an unknown quantity, at the wrong end of the bridge; and it would probably not have been discovered had Coney Island not been at the other end of it! Queens and Richmond were foreign countries, populated by enemy aliens; and The Bronx—what a place that was!—a vast and impenetrable wilderness beyond the Harlem River, with a total

population of only 200,000 scattered throughout its vast acres. Today they crowd more than that into a single Bronx express!

The auto was as scarce then as steak is on a banquet menu today, and you had to wait until tomorrow to learn what Union was on strike today instead of hearing it over the radio tonight.

Unknown, and almost undreamed of, were the oil burner, the household refrigerator, the washing machine, and all the other labor-saving gadgets that have so reduced the mechanics of housekeeping that it has freed the women to spend their time at the movies and to raise hell in the night clubs.

Life was simple and peaceful then. There were no night clubs as we know them now but Weber and Fields rent the air with their raucous comedy. New York was a city of horse cars and hansom cabs, of mustached policemen with iron hats and bewhiskered politicians with iron heads. Those were the days when men were men and wore whiskers—and women were women and wore corsets.

The carriage trade patronized the opera and lived in the brownstone fronts off Fifth Avenue, where, to the disgust of the Great Unwashed, it was rumored they had moved their bathrooms in from the back yard and located them next to the front parlor. In the hinterlands, however, Chic Sale still carried on his lucrative profession.

John McCormack was then the idol of the music-loving public and thrilled great audiences in the concert halls of every city with his rich tenor voice. Today Frank Sinatra in a voice that sounds like the wail of a lost soul croons and sends the jitterbugs into a series of convulsive antics that have all the artistry of a race riot.

Then the young folks found their pleasure in visiting each other's homes. There, gathered around the old square piano, they filled the house with the melody of such songs as "I'll Take You Home Again, Kathleen." Today they gather around a juke box in some joint, roaring loudly, "Lay That Pistol Down, Babe—Lay that Pistol Down."

The pessimists may look with sorrow on these things and regard them as signs and portents of a world gone mad and of a civilization that is again passing through the twilight of an approaching Dark Age, but the optimists will consider hopefully that it is merely a passing phase—providing the necessary outlet for the emotions that will lift the human race above the tragedies and errors with which the world is beset today.

Certain it is that we need not look far to find examples of human behavior that will far outweigh the trivialities of which I have spoken. The shallowness of the few only accentuates the inherent soundness of the many, and there is hope in a world in which men in endless numbers will face death and agony in every form in defense of their homes, their liberties, and their sacred honor.

We have in our Society two charter members still living. One of them is Mr. Charles F. Hauss. He is not well known to most of us because most of his career has been spent in the countries of Europe and Asia, representing American companies. The other one is Mr. Homer Addams, who I regret to say, had fully expected to be with us and is in town, but, unfortunately, is laid up at the Engineers' Club, suffering from a very heavy cold, and has had instructions from his doctor not to leave his room. We are certainly sorry that Mr. Addams is not with us. He is one of our very distinguished members, a Past President of this organization, who has served faithfully and continuously for the fifty years of the Society's existence. Only two weeks ago he appeared before the New York Chapter and gave them a very memorable talk, reminiscing on some of the days of the past. We are sending a message to him tonight, from the Past Presidents assembled here, of whom there are 19 registered at this meeting. I think it is an indication of what this Society means to men who have served it, when they continue to serve it right up until the very end.

#### I.H.V.E. MEDAL

We will now proceed with the presentations. There have always been close ties of friendship between this Society and the *British Institution of Heating and Ventilating Engineers*. The members have visited back and forth and both Societies have had a keen interest in each other's doings. This year the British Institution has signally

honored one of our members by awarding him a silver medal, and this has been selected as an appropriate occasion for its presentation. The presentation will be made by the man who has so capably served as our President for the past year, and who will now be introduced to you publicly for the first time as Past President Merrill F. Blankin.

MR. BLANKIN: Toastmaster, Mr. President, Ladies and Gentlemen: This is an unusual honor tonight, to be able to represent the *Institution of Heating and Ventilating Engineers* of England. It is also unusual tonight that we have two medals to present, and this first one is one that is awarded about every five years by our companion society in England. The Council of the *Institution of Heating and Ventilating Engineers*, on October 12, 1943, voted their Silver Medal to one of our outstanding members, Thomas Chester, Detroit. The medal was presented for the best paper presented before them during the 5-year period of 1937 to 1942. They were quite anxious to have this medal formally presented to Mr. Chester, but, of course, due to present war conditions they could not arrange to do so. They asked us to act in their behalf and forwarded the medal.

The paper that Mr. Chester presented was on the subject of *Drying by Heated Air*, at the time he was consulting engineer for the British War Office, in charge of the design and installation of an extremely large air conditioning system for the dry underground storage of munitions in mines and quarries. Mr. Chester was in London at the time of the outbreak of the war in 1939 and spent four months there during the war. He has since been in the United States, doing important work as a civilian engineer for the U. S. Army Engineers.

On behalf of President Crittall, of the *Institution of Heating and Ventilating Engineers* of England, it gives me profound pleasure to present the Silver Medal of that Institution to you. May I congratulate you Mr. Chester?

THOMAS CHESTER: Mr. Ex-President, Ladies and Gentlemen: After five years it is quite a surprise to me to receive a medal. The worst thing about getting a medal, I find, is that there are no roses without a lot of thorns. And the thorns consist of remarks by all and sundry as to why I was awarded this medal! I am trying to forget most of them as rapidly as possible, but two stick in my mind: one is that this was for "bravery behind the lines"; and the second was that I was awarded the medal on account of being put out of England to save food!

#### F. PAUL ANDERSON MEDAL

TOASTMASTER DRISCOLL: Mr. Blankin will now make the presentation of the F. Paul Anderson medal, which is awarded to the person nominated by a special committee for his outstanding service to the arts and sciences in which we are specifically interested.

MR. BLANKIN: It is quite fitting that the F. Paul Anderson medal should be presented tonight to Dr. Ferry C. Houghten. He has rendered to this Society a most distinguished service. He was Secretary of the Society in 1924 and 1925, at which time he became Director of our Research Laboratory, and remained as director until 1942, when he changed his uniform. He took on the two and a half stripes of a lieutenant-commander in the United States Navy, and has been serving the United States since that time.

He has been doing research work in the air conditioning section of the Bureau of Ships, and the Research Division of the Bureau of Medicine and Surgery. I cannot take time to tell you all of his accomplishments. They are well known. But I am very much interested to find that three of his sons are in the service of the United States, as is also his son-in-law.

And so, Commander Houghten, it is indeed a privilege and a pleasure to present to you the F. Paul Anderson Medal, which you have so nobly earned.

LIEUTENANT COMMANDER HOUGHTEN: President Blankin, Members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and Guests: Words fail me to adequately express my appreciation for this award and this honor, other than to say a plain "I thank you."

I am reminded, however, that honors, acclaim, and position in life rarely come to a man as the result of his own efforts. More often they come as the result of opportunities given by his friends, by help and cooperation from them; and in this case that is doubly true. My opportunities to serve were the problems that you gave me, the problems which the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS gave me. While still a young lad at the U. S. Bureau of Mines, John R. Allen, first Director of the Society's Laboratory, took me into your Laboratory. Later it was my privilege to work under Director Scipio and that great leader of men, Director F. Paul Anderson, whose name and whose likeness appear on this medal.

I worked at your problems at the Laboratory with the members of the staff, who, likewise, helped to earn this medal. It was my opportunity to work with many chairmen of the Committee on Research—noble men, every one of them, the members of the Committee on Research, the chairmen and members of the Technical Advisory Committees, and at last every member of the Society. It was these opportunities, it was the work of each and every one of these men who shared the responsibility of doing the things for which I get this medal, and I am very mindful of the fact that, while it is my privilege to receive it, to hold it in esteem, as I always will, as the greatest honor that has ever come to me, I shall always feel, nevertheless, that I am carrying it also for those other people.

#### PAST PRESIDENT'S EMBLEM

TOASTMASTER DRISCOLL: The previous presentations were of a special nature. The next is the annual award of the Past President's emblem to the retiring President.

It is singularly appropriate, it seems to me, that this presentation be made by a man who preceded Mr. Blankin in that high office, and who served with marked distinction and success, and who himself is now a most worthy member of that august group known as the Advisory Council, whose advice is never taken but which is composed of all living past presidents of the Society.

Hailing from the distant City of Seattle, Wash., he has been a most faithful attendant at our meetings and an active and valuable member of our Society, and I am happy to present to you at this time Prof. E. O. Eastwood.

PROFESSOR EASTWOOD: Mr. Toastmaster, Ladies and Gentlemen: I think we may well be proud of the opportunity tonight to honor another Past President of the Society. The Chairman has told you that there are 19 of the 23 living Past Presidents of the Society in attendance at this 50th Annual Meeting. All these men have served the Society well. You know the policy of the Society, the policy of the award sometimes indicated by appointments to committees, sometimes by election to council or to administrative duties.

During the past 50 years 49 other members than Mr. Blankin have received this honor, but not until the year 1924 was the award made in the form that it is today. Few of you have the opportunity of seeing this medal. It is a gold medal, suitably inscribed on the back in the form of the emblem of the Society, and mounted with five diamonds. Those who have received it have reflected not only upon themselves but also upon this Society, and no one, in my opinion, has earned this medal by self-development more than has Merrill F. Blankin.

When he was elected last year Mr. Blankin did not accept this as the culmination of a deserved honor but as an opportunity to go forward, to go farther with the work which he had been doing for years for this Society. He is gifted with a pleasing personality, and, with that pleasing personality and a cheerful disposition, he has gone in an ingratiating way out to the various sections of the Society, and he has substantially increased the membership of the Society. During his administration several of the most important branches of the Society's work have been developed, notably that of the Research division. It has been enhanced in interest and in products, and this work will continue to go on as it has before, with increased effect.

Mr. Blankin has also served as treasurer of the Society. He has served on the Council and he is entering the ranks now as past president.

Mr. Blankin, for all these reasons I take pleasure in presenting to you, in the name of the Society, this Past President's emblem, in appreciation of the services

you have rendered, and in token of the high esteem in which the membership holds you.

MR. BLANKIN: Thank you, Professor Eastwood. I am sure I do not deserve all the kind words that have been said about me. True, we have accomplished a great deal this year. True, possibly, that I have worked hard and traveled far—but nothing could have been accomplished without the complete cooperation of everyone, and I have certainly had that complete cooperation.

I suppose I should be a little sad tonight, to see 25 years of work for the Society ended. I am told that I must go into that Advisory Council, that does not advise, and that does nothing. But perhaps I am a little younger than some of those who have approached that bench on the sidelines.

I am happy that I am going to be able to be of further service to the Society during the coming year. I am not sad; I am happy. I am happy to see my good friend J. H. Walker down here tonight, looking so hale and hearty. I am happy to see a banquet tonight where we have over 800 in attendance. I am glad to see a registration of nearly 1000 at this meeting. I am glad to have had the opportunity to be present at the Society's 50th Annual Meeting.

I believe I am the first native-born Philadelphian to have been president of the Society. I am happy to have represented Philadelphia. I am happy to see the foundation laid this year for the further progress of the Society, and I think we are going forward to that greater future that we are predicting.

I joined the Society 25 years ago. At that time our annual income was \$11,000. Our income today is nearly \$150,000 a year. What will it be 25 years from now? I hope that I can be with you at that time, and I hope that I can continue to serve the Society down through the years.

All I can say now is that I wish you God-speed—and let's all work together next year.

TOASTMASTER DRISCOLL: The guest speaker of the evening is probably known to many of the audience because of his many forceful speeches over the radio and from many platforms—Dr. Daniel A. Poling.\* Dr. Poling is a minister, a leader in civic movements, an editor, radio speaker, and an active worker in both the first World War and the present conflict. During the past year Dr. Poling has visited all of the war fronts, with the exception of Australia, and is familiar with the troops in all of these theaters of war.

Dr. Poling's subject was "The War on Four Fronts," and he gave a vivid description of his experiences of seven months overseas and contrasted some of the factors that made the present war a new kind of war as compared with World War I.

#### THE WAR ON FOUR FRONTS

DR. POLING: We are winning the war but the war is not won, and until it is won it could be lost.

Of the last twelve months I have spent nearly seven overseas, and in that time have visited all the active fronts, but not all the sectors in England, Africa and Asia.

I tell you tonight that the war will not be lost by the armies of the land and by the ships of the sea and the air. It will not be lost by our sons over there. That issue will be decided by us over here. The war will not be won in the Year of Our Lord 1944—nor ever—by captious criticism of leadership, by slow-downs and strikes and lockouts; nor by any plan of appeasement or selfish nationalism—not now, or ever. Now minutes are men. Presently they may be your man.

Two years ago the President of the United States said, "This is a new kind of war, and it is a new kind of war because ours is a new world."

There appeared recently a book which is of interest describing the journey of a former Philadelphian from Philadelphia to Williamsburg, Va. He was eight days in the saddle and he had some strenuous adventures.

\* Pastor of Baptist Temple and a Major in the Army Chaplain Corps.

In December I spent eight days flying back from Chungking to Philadelphia, and of the eight days I was on the ground three; yet I traveled 16,000 miles.

This is the new world in which we live. I crossed the Atlantic Ocean first in December, 1917, when many others were crossing. I spent 13 days at sea on the old New York of the United States line. I saw plenty of ocean. I flew the Atlantic first in September, 1941, before Pearl Harbor, without even seeing the water. We took off one night from Gander, in Newfoundland, went up through the fog and mists that came booming down the Straits of Labrador, came out at 17,000 ft under the stars, stayed there for 9 hours and 10 min, and dropped down through fog and mist to a happy landing in Scotland. We had not seen the water.

Lt. Reynolds stepped from the B-24, stretched his cramped legs, and said, "I have enough gas to get back to Chicago." I think that was an exaggeration, but he had gas enough to get home.

Twenty-five years ago, when some of those of us who are now venerable men, watching our arteries, were younger, the oceans divided the continents. Now they do not interrupt our passage over them. Twenty-five years ago the ether above us was friendly. Now it is a sea filled with potential fleets of death and destruction.

It is a new kind of war because it is a new world. Years ago I read an article in the Scientific American which said that the time would come when a man could ground his instrument and talk to his friend who might be lost in a jungle of Africa. I didn't know what it meant to *ground* an instrument, and I didn't believe it, anyhow; but I have lived to do just that.

In 1938 I found myself in Honolulu, visiting a friend who is a short-wave enthusiast, and one evening he introduced me to a friend of his whom he has never seen, who has a great sheep station in Northern Rhodesia, in Africa, and he said to his friend as they sat in conversation, "My friend, Dan Poling, is coming out there in a few months, and I hope you two men meet." It was as simple as that. A few months later I came down to Bulawayo, the capital of Southern Rhodesia, from Victoria Falls, and the gentleman met me and took me out to World's End, where I saw the grave of Cecil Rhodes, then took me to his station, and that night I talked to Philadelphia. The arrangements had been perfected, and I whispered across the world, or through it, or around it; I talked on Friday and was heard on Thursday. The international date line took care of that!

Yes, this is a new kind of war because we live in a new kind of world. Last March I spent two weeks in the Midlands with some of our bomber squadrons. I was preparing some short articles then for the Philadelphia Enquirer. I called them, "A Preacher Looks at War," and I saw plenty. I was with the briefings in the morning, and then I would stand with the chaplains of the two faiths while the great ships would take off the ground—crews waiting on the field. When the time had elapsed, we would go to greet them as they returned.

One evening—it was the evening of the day when we suffered our heaviest bomber losses up to that time—when the great ships came in I found myself afterward at headquarters. A lieutenant-colonel who had been on the mission, with adhesive across his brow (he had received just a scratch from the flak), was sitting tense and drawn at his desk. In front of him was the dispatch telling of the Boeing strike in Seattle. It was not a serious matter; the men were not out long. Lt.-Col. Wade turned to me and said, "That, sir, is the black damp of death on this wing. If the men at home would see not seven ships down, but seventy men, every man irreplaceable, it would be different, I think."

A ground sergeant came in presently and stood at the desk. Until I die I shall remember him. He was crying like a hurt child. He said, "Sir, I have two brothers in the Boeing works in Seattle. They wouldn't do that to us; they don't understand. They are not to blame."

The boy who was crying at his commander's desk defended his brothers who did not understand!

There is vastly more than first appears behind a strike, gentlemen. When I was a little boy in Cambria, Pennsylvania, I saw bodies carried out from the mouth of the J. C. Stimmer mine. There had been an explosion—black damp; and ever since that day I have seen the red in the black of the coal.

I don't believe any man has ever been paid too much to bring coal out of the depths of the earth. But men out there now can accept no reason for a strike anywhere because, you see, it is life and death for them and for the cause they represent, and that, to them, is all that matters. No slowdown, no strikes. That applies to all of us. When word comes of inferior materials manufactured at home they stand literally aghast. They can't understand it. And may I say this: I do not believe there is a man in any foxhole—I do not believe there is a man in any Flying Fortress—I do not believe there is a man in any submarine who does not believe that we should establish on the home front the universal service principle.

To me the question, will it prevent strikes? is absolutely incidental. That issue will be decided over here. The principle stands. We are all in; if freedom wins, all are free; and if freedom loses, then all have lost.

Strangely enough, the text of the preacher has in our time been fulfilled. We cannot win with freedom in this war without winning freedom at least for our foes, as well as for our friends. For I tell you that never again can the United States of America enjoy the justified hope of an ordered freedom with the justified hope that war itself shall come not to our children's children unless the last man of the last country enjoys that hope—for it is indeed a new kind of world.

We live too close together; the most remote tribe is now in our back yard.

I have been interested in our bond campaign in Philadelphia. I have been serving, representing my particular faith on the committee, and have been calling on some of my brethren. One of these men told me that he was not interested in the proposition because he did not believe in war, wouldn't touch it, and wouldn't run the risk of touching it.

"Well," I said, "that is all right. But aren't you in favor of feeding the men who are over there? Won't you buy a bond and be interested in buying a bond to feed the men who are over there? Bonds buy bread as well as bullets."

I told him that last April, after Rommel's break through in Tunisia—and I was in North Africa and in Tunisia at the time—there were nearly 80 hours when the men of the 34th Division in advance positions had only tomato juice and "iron bread," hardtack, and they didn't complain, because they knew that other men were dying to open the roads to get the food to them. So I said to this man, "Don't you think that you would be willing to buy bonds to get food to the men out there? You see, it is like that."

We pay taxes, and I submit to you tonight that by every sacred test, the last man, whatever his convictions may have been before the theory became a fact, and the battle is joined, the last man at home has responsibility before God and man equal with the responsibility of the man in the Flying Fortress and in the foxhole.

I have a lad, second officer on the SC-504, somewhere in the Southern Pacific, perhaps covering landings in the Marshalls now. And another at the controls of a great bomber somewhere, if he remains alive tonight. They are there and I am here. But before the Father of us all, I feed the shells into their cannon and the gas into their tanks, and morally I am responsible with them, though they eventually have blood upon their hands perhaps.

There is only one way to stay out of war, and that is to keep it out of the world and away from every human front. That is the realism of this war.

There is something more that I would say, for tonight I have a sense of mission. I would speak, though faintly, for those who are not present and are not able to articulate their sentiments.

I do not believe there is a man in a foxhole nor in a Flying Fortress nor in a home camp who does not feel that this great, free America should make it possible for every man in uniform to cast his ballot. There is no state right, there is no party right, there are no other rights that should deprive the man in uniform of his voting rights or that should jeopardize his voting rights. I do not think the man in uniform would have thought about it at all if the question had not been raised. When the election took place in Philadelphia this fall I was in Base Hospital 20 on the Ledo Road, out in front of Assam, one of the largest hospitals in the Army—2500 beds. When I came off the trail into that ward I looked up into the eyes of a girl from my own congregation. It was ten days after the election before it occurred to me that there had been an election in Philadelphia. I was

deeply interested in that election. Then I asked the great surgeon, Dr. Rovdin of Philadelphia, whether he knew how the election had gone, and he said, "What election?"

I said, "Philadelphia!"

It was another ten days before we found out. They had not been thinking about elections. But now that the question has been raised, believe me, they are thinking. I had a letter from a lad named Joseph Engelhart, an office boy, a graduate of Girard College. He wrote and said, "You must have brushed me when you passed at headquarters, for I was standing post right there." He said, "I cried when I found you had been so close." But you do not easily recognize each other. That was the day that Faïd Pass was retaken. Then he said something more. He described the battalion left on the mountain when Rommel broke through. That first night, you may recall, we lost a battalion and a half, as prisoners of war. But this battalion decided to make the effort to get through and they went through eight miles of enemy lines to rejoin their division, the 34th. He said, "On the last day on the mountain (and may this be my tribute to the chaplains in this war) we could not go to church; there could be no services. We were in our foxholes and we were strafed all day and there was an icy rain. Then the chaplain carried marked testaments to us in the foxholes and told us to read the verses and pass them to the next foxhole, and so the church came out to us. That night we started back. The chaplains stayed on the mountain with our wounded and with the German wounded." But those words, "the church came out to us," that is the challenge to the church of your faith and mine today—"the church came out to us."

I found religion on all these fronts, but it did not follow the usual patterns. There is mighty little time for men to sing hymns in front of an altar, but I did find religion pure and undefiled—often obscured by heavy oaths, to be sure, and obscenities and sex—so that if you heard only the surface sounds you might reach the conclusion that the army is pagan, but you must listen, as I listened, and hear the deeper note of character beneath—religion pure and undefiled. The Major with tubercular kidneys, going home to die, rubbing the backs of men acutely ill; the driver on the mail truck on the Ledo Road who had a cough that sounded like pneumonia but who drove twelve miles out of his way to get a lost kid from Georgia back to his outfit; the nurse at Marker 31, hard on the boundary of Burma, who was pinning a curtain on her window to make it look like home to homesick men—religion pure and undefiled.

I could not tell you what they think, for who knows what a man thinks? But they think and they act. It was in Base Hospital 38 one Monday morning that Brigadier-General Chieves, Chief of Staff to General Royce, gave me the high privilege of pinning on the ribbons while he read the citations—22 men from the beaches of Salerno and from Italy. That hospital was a poignant sight. Men with their legs toward the ceiling, held by weights in that position; men between sandbags; men terribly burned; silent men on beds, their eyes covered—eyes that would not open again. The General read the citations and I pinned on the ribbons. When we came through the entrance and were leaving, the registrar, Medical Major Glen Williams, said, "General, there is an unannounced item on the program, if you are willing." The General said, "All right, what is it?" The Major said, "The men want to sing."

There were three instruments—a banjo, a violin, and a mouth organ—and so they sang. The wounded from Salerno and from Italy, the maimed and the blind, sang "God Bless America."

"God bless America, land that I love—  
Stand beside her and guide her  
Through the night with a light from above."

I wish that every man in America could have stood as I stood there to hear them sing. I didn't sing. I had not earned the right to sing in that choir.

They are fighting to get home. More often than in any other phrase do they express it thus—fighting to get home.

But there is vastly more than that in it. There isn't an isolationist in uniform whom I have met. They are not imperialists. They do not want a single inch of

the ground their feet have trod. But one thing they know, that since we could not stay out of the war, we are fools to try to stay out of the peace.

They are realists; they are fighting to get home. But what home? They are fighting with the high purpose that they shall have their part in winning a decent peace. They believe that winning the war and winning the peace are one and indivisible, and that to win the chance to build a decent world we must first of all win the war. The peacemakers of this night, men and women, are the men in the camps—your men and mine, and the men on the sea and on the land and in the air who are fighting to help win the war—that is the realism of the war.

I wish that I might share with you certain Chinese impressions because when our men come back they are likely to be sadly disillusioned. The Chinese situation is significantly more than the visit of the radiant Madame Chiang Kai Shek. Some things are difficult to understand. I had three days with a Colonel, our field artillery expert, who is with the advance Chinese combat divisions on the Burma frontier on the China side. He was recovering from four wounds and was just out of the hospital. Any one of the wounds might have been fatal, and they were received from Chinese bandits. He was in command of a small convoy of three trucks, five other officers and nine enlisted men, on his way to the front, 90 miles out of Kungming, when, rounding a curve, they found themselves in the midst of bandits who were looting three Chinese trucks. Fire opened at once. All the officers were wounded, the Colonel seriously. They got two of the trucks back to the Chinese town at the rear, just around the curve. The Colonel said, "It is tough to be out here fighting in a common cause and then to be shot up like that. But we need to remember that China is now passing through what was our frontier period of half a century ago. We need to remember that when Chiang Kai Shek first faced the Japanese he controlled only seven provinces of China and had civil wars of one size or another confronting him in every other province; that he stood between the extreme left and the extreme right. You must remember that," he said.

We must not forget that there was a time when in our own great Western territories and states, ranges were rustled and burned, communities were terrorized, stage coaches were robbed. Then came the Vigilantes and the United States marshals, and gradually law and order emerged. Well, that is China of today. But let us never forget that this great leader, incorruptible, has been able to stand through seven long years, refusing to sell out to the highest bidder when Japan, to save face now, would retire with scarcely a memory of what she first asked for.

Through the complexities of these war days, the courage of China through these seven years shines like a star resplendent on the field of international affairs.

There are a number of things that stand clear for us, that reach us tonight in this great room. But, above all things else—and with that I am sure I should make way for the program that is to follow—is this compulsion to national unity—unity, not uniformity. We be Catholic and Protestant and Jew, we are of all faiths and all colors. In this great national stream there are the blendings of all the racial currents of the earth, but it was our great one who said that "united we stand, divided we fall." This is the test at last, not of our ability to meet the foreign invader, but of our ability to face and meet ourselves. I do not finally fear today any foe that might embark from foreign shores to seek us out and do us hurt. The only thing that could destroy us would be this thing, this dry rot from within. Out there there is unity. The tremendous significance of the task itself is ever before men, but here it seems more difficult for us to understand, to understand each other.

I have been with the First Division in the other war and with the Twenty-sixth Regiment, so I was particularly anxious to see the First Division in this war. I found it 12 miles in front of El Guettar on the Gafsa Road. It was the day the great tank battle opened. The tanks came down between the 16th and the 26th, moved beyond the guns and fanned out left and right across the desert. That night advance patrols from the First Division and the Ninth met the British Eighth Army, Montgomery's desert army. Until the end of time there will be discussions and debates as to who it was, and the exact spot! Naturally, I was anxious to see my successor, after 25 years, and I found him, Chaplain Chase, who had been cited, decorated, and promoted on the field of battle.

Teddy Roosevelt told a story about him that Chaplain Chase says is apocryphal, but it is a good story and I brought it back, much to the confusion of the Chaplain.

He was called "young Teddy" 25 years ago. He is 25 years older now. I found him the second in command of the First Division at headquarters in a little oasis, Major-General Terry Allen in command. I had an afternoon and a night there and we talked about other times. I had seen him as a second lieutenant and a first lieutenant and a captain and a major and a lieutenant-colonel and then colonel. Now as a brigadier-general he was back with his old organization again. I told him that I was looking for Chase, and he said, "That man is absolutely a fool," and he described the days when Rommel was moving through the lines of the 34th in the north and the First Division was outflanked. He said, "We were moving back, getting our vehicles out, moving the 77th Evacuation Hospital. We were doing pretty well, and the orders were for men to get out of their vehicles when the strafing planes came over, for them to get into the fields and then, when the raids were over, to get back into their vehicles as quickly as possible. We had things well in hand. Four o'clock in the afternoon I heard a jeep struggling up the road. I rushed out and threw up my hand. All other vehicles had stopped except this jeep that was struggling up the road. I jumped on the running board. It was the Chaplain. He had his foot right down to the board. I called to him but he did not look up. He said, 'Sir, I have waited six months for this jeep and I am not leaving it behind now.' Then he motioned to the rear, and I saw that he had two wounded men in the rear of the jeep, and I got off the running board."

Well, I found Chaplain Chase at the graveyard behind corps headquarters. With him I found two other men, MacEvoy, the priest who for eight years was pastor of the Floral Park Parish on Long Island, and Stone, the Jew. I found them burying their dead, eleven white bags by the road—Arabs digging in the sun-baked sand, going down less than four feet. That was deep enough. When the graves were ready we lifted the sacred burdens and placed them in their last resting place. Then Stone read above his Jewish dead, and MacEvoy read above his Catholic dead, and Chase read above his Protestant dead, but they read from the same Book and they prayed to the same God.

It is like that, men and women, in this war. It is a unity that transcends differences. It is a unity that strengthens every worthy individual loyalty. The Catholic is a better Catholic, and the Jew is a better Jew, and the Protestant is a better Protestant for being an American—we are better for being *Americans all* in a time like this.

The 2nd of February, one year ago tonight, the *Dorchester*, a small cargo transport, was moving steadily through iceberg-infested and submarine-crowded waters—one year ago tonight. At 1:15 the torpedo struck, almost amidships, and silenced the engines. The ship listed and in less than 25 min went down. The story runs that there were on board that ship four chaplains—a Catholic, a Jew, and two Protestants. They were friends—intimate friends. They had on their life belts. They did their best to quiet panic. Six hundred and seventy-eight men of the 904 on board the *Dorchester* died that night. Then, when there was nothing else to do, these four men took their life-belts and forced them upon four enlisted men, and gave away their chance to live.

I went out to Valley Forge in the spring after my return from my first overseas visit of the past year to talk to Grady Clark, an engineer of the *Dorchester*. He told me what happened on the deck and how, as he slid under the rail and swam a bit away, he saw the prow come up high and the ship slide under. He said, "Side by side, the four chaplains stood on the deck, praying still for the men who were struggling in the water."

One of these men was Washington, the priest; another was Goode, the Jew; another was Fox, the Methodist, and the fourth, the youngest, was our son. This is the anniversary that I am celebrating. I have come here tonight to say to you that the only justification for my presence is the tremendous sense of privilege that I feel, as a Protestant clergyman, in adding my word to the high calling of American citizenship and to this unity in which we shall be free, in which we shall find the answers, in which we shall preserve and perfect liberty, not only for ourselves and for our children, but for all who shall come after us.

"O Captain, my Captain—

The night is black with thunder.

The ship is tossing in the waves' wild will.

The foaming whitecaps hiss and slip from under.

The mighty engine's throbbing heart is still.

"Where are we drifting in this night of peril

Hysterical pilots wrangle at the wheel.

With rudder gone into the glooming terror,

The great ship rushes on uneven keel!

"O, ye of little faith, am I not able—  
I hear my Captain's answer brave and strong—  
'When every anchor slips her faithless cable  
To guide the ship that I have kept so long?"

"The little pilots at each other railing  
Soon fall asleep, and all their strife forget.  
But I, who set the fleets of time to sailing,  
Have this great nation in My guidance yet."

God bless you, every one.

TOASTMASTER DRISCOLL: Dr. Poling, I am sure that I echo the sentiments that are in the minds and hearts of all here tonight, of their great appreciation for this fine, vital, vigorous talk of yours. I am sure that as long as any man or woman in this audience attends meetings of this Society, they will recall this talk as one of the highlights in the entire history of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. We thank you.

It would seem almost sacrilegious now to tell you what is on our program. I am going to just let you rest a moment, while I take a moment of your time to read a few lines that seem appropriate to an occasion like this, that I clipped rather unexpectedly from the paper a day or two ago:

"So long as there are homes to which  
Men turn at close of day,  
So long as there are homes where children are,  
Where women stay—  
If love and loyalty and faith be found across  
those sills—  
A stricken nation can recover from its gravest ills.

"So long as there are homes where fires burn,  
And there is bread—  
So long as there are homes where lamps are lit,  
and prayers are said—  
Although a people falter through the dark,  
and nations drop—  
With God Himself behind these little homes,  
We sure have hope."

## PROGRAM 50TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

HOTEL PENNSYLVANIA, NEW YORK

JANUARY 30-31, FEBRUARY 1-2, 1944

*Sunday, January 30*

1:00 P.M. Registration (Ballroom Foyer)

*Monday, January 31*

8:30 A.M. Registration (Ballroom Foyer)

9:30 A.M. BUSINESS SESSION (Georgian Room)  
M. F. Blankin, *President*, presiding

Greetings

Anniversary Messages

Introduction of Guests

Reports of Officers and Council

Report of Chapter Development Committee, W. A. Russell, *Chairman*

Report of Committee on Research, C. M. Ashley, *Chairman*

Report of Guide Publication Committee, P. D. Close, *Chairman*

Amendments to Research Regulations

Report of Tellers of Election, R. A. Wasson, *Chairman*

- 10:30 A.M. Ladies Visit to Statler Research Kitchen (brief talk on Hotel Wartime Food Problems)
- 12:30 P.M. Get-together Luncheon (Grand Ballroom)  
 A "Welcome to New York" luncheon for all members and ladies—  
*Toastmaster:* Homer Addams; *Speaker:* Honorable Fiorello H. La Guardia, Mayor of City of New York. *Subject:* The Post-War Era Offers a Challenge to Engineering
- 2:00 P.M. TECHNICAL SESSION (Georgian Room)  
 S. H. Downs, *1st Vice-President*, presiding  
 The Theories and Practices of the Past Fifty Years in Heating and Ventilation, by S. R. Lewis  
 Review of the Development of Standards for Comfort Air Conditioning, by Lt.-Comdr. F. C. Houghten  
 PANEL DISCUSSION—The Future Trends of Heating, Ventilating and Air Conditioning—Dr. B. M. Woods, *Chairman*
- 2:00 P.M. Ladies Card Party and Tea (Parlor 2)
- 6:30 P.M. Past Presidents' Dinner (Parlor A)
- 8:00 P.M. Tomorrow's Uses of Radiant Energy, by Samuel G. Hibben—Salle Moderne
- 9:00 P.M. Old Timers Night (Roof Garden)  
 An informal party for Members and Ladies with a real Gay Nineties flavor; Zeke Lockwood, Master of Ceremonies.

*Tuesday, February 1*

- 9:30 A.M. TECHNICAL SESSION (Georgian Room)  
 C-E. A. Winslow, *2nd Vice-President*, presiding  
 A Study of Intermittent Heating of Churches, by F. E. Giesecke  
 The Resistance to Heat Flow Through Finned Tubing, by W. H. Carrier and S. W. Anderson  
 The Control of Air Streams in Large Spaces, by G. L. Tuve and G. B. Priester
- 10:00 A.M. Ladies Visit to American Museum of Natural History Exhibit of Wild Animals in Natural Habitats followed by Luncheon and at 2:00 p.m. attend show at Hayden Planetarium—The Story of the Earth.
- 11:45 A.M. Leave Hotel Pennsylvania for Inspection of Mechanical Equipment of Radio City
- 2:00 P.M. TECHNICAL SESSION (Georgian Room)  
 S. H. Downs, *1st Vice-President*, presiding  
 Discoloration Methods of Rating Air Filters, by F. B. Rowley and R. C. Jordan  
 The Axial Flow Fan and Its Place in Ventilation, by W. R. Heath and A. E. Criqui  
 The Aerodynamic Development of Axial Flow Fans, by T. H. Troller

*Wednesday, February 2*

- 9:30 A.M. TECHNICAL SESSION (Georgian Room)  
 M. F. Blankin, *President*, presiding  
 Address: Fuel Economy and Army Heating, by Major Arthur W. Nelson, C.E.  
 Address: Collective Bargaining for Engineering Employees, by George T. Seabury, *Secretary, ASCE*
- 11:00 A.M. Inspection Trip for Ladies of famous William Randolph Hearst art collection at Gimbels

2:00 P.M. TECHNICAL SESSION (Georgian Room)

M. F. Blankin, *President*, presiding

Optimum Surface Distribution in Panel Heating and Cooling Systems,  
by B. F. Raber and F. W. Hutchinson

Unfinished Business

New Business—Resolutions

Installation of Officers

Adjournment

7:00 P.M. ANNUAL BANQUET (Grand Ballroom)

Toastmaster—W. H. Driscoll

Speaker—Dr. Daniel A. Poling

Subject—The War on Four Fronts

Presentation of Silver Medal of the Institution of Heating and Ventili-  
ating Engineers (London) to Thomas Chester, Detroit, by Pres.

M. F. Blankin

Presentation of F. Paul Anderson Medal to Lt. Comdr. Ferry C.  
Houghten, by Pres. M. F. Blankin

Presentation of Past Presidents' Emblem to M. F. Blankin, by  
Prof. E. O. Eastwood

COMMITTEE ON ARRANGEMENTS

ALFRED J. OFFNER, *General Chairman*

*Honorary Chairmen*

HOMER ADDAMS, W. H. CARRIER, W. H. DRISCOLL, W. L. FLEISHER, D. D. KIMBALL

*Vice-Chairmen*

R. H. CARPENTER, J. C. FITTS, W. E. HEIBEL, C. S. KOEHLER, A. E. STACEY, JR.,  
R. A. WASSON

*Banquet*—ALFRED ENGLE, *Chairman*;  
RUSSELL DONNELLY, *Vice-Chairman*;  
E. E. ASHLEY, R. W. CUMMING, M. C.  
GIANNINI, JOSEPH WHEELER, JR.

MRS. A. V. HUTCHINSON, MRS. C. S.  
KOEHLER, MRS. G. E. OLSEN, MRS. R.  
A. WASSON

*Finance*—W. M. HEEBNER, *Chairman*;  
H. W. FIEDLER, *Vice-Chairman*; R. R.  
FERGUSON, G. E. OLSEN

*N. Y. Chapter Special Committee*—C. F.  
ROTH, *Chairman*; J. G. EADIE, J. C.  
FITTS, C. S. HOFFMAN, H. C. MEYER,  
JR., G. M. SCOTT

*Hospitality*—H. S. WHEELER, *Chairman*;  
W. J. OLVANY, *Vice-Chairman*; T. N.  
ADLAM, F. E. W. BEEBE, H. H. BOND,  
ERNEST GRABER, E. B. JOHNSON, C. A.  
MILLER, L. L. MUNIER, J. R. MURPHY,  
H. B. EELLS, JOHN WHITE, W. J.  
OSBORN, V. F. SELF, J. H. PFUHLER

*Publicity*—R. V. SAWHILL, *Chairman*;  
CLIFFORD STROCK, *Vice-Chairman*; A.  
A. BEARMAN, C. H. B. HOTCHKISS, O.  
O. OAKS, W. J. OSBORN

*Inspection*—C. S. PABST, *Chairman*; P. B.  
GORDON, *Vice-Chairman*; H. L. ALT,  
J. A. HELLER, M. H. HIRSCH, F. D.  
LAWRENCE, RUDOLPH POLLAK

*Sessions*—E. J. RITCHIE, *Chairman*; C.  
H. FLINK, *Vice-Chairman*; E. E.  
ADAMS, S. R. APT, THOMAS BAKER,  
P. G. GRIESS, G. D. GULER, C. R.  
HIERS, H. P. WAECHTER

*Ladies*—MR. AND MRS. H. J. RYAN, *Co-  
Chairmen*; H. S. JOHNSON, MRS. R. H.  
CARPENTER, MRS. H. W. FIEDLER, MRS.  
W. L. FLEISHER, MRS. O. E. FRANK,

*Special Events*—A. C. BUENSOD, *Chair-  
man*; A. F. HINRICHSSEN, *Vice-Chair-  
man*; V. J. CUCCI, C. A. FULLER, F. D.  
McCANN, M. F. RATHER, H. J. ROSE,  
W. A. SHERBROOKE

## GREETINGS ON SOCIETY'S 50th ANNIVERSARY

AMERICAN SOCIETY OF CIVIL ENGINEERS, by Malcolm Pirnie, *President*, and George T. Seabury, *Secretary*: On this memorable occasion of the Golden Anniversary of your Society whose members have multiplied the comforts, advanced the health and improved the efficiency of the citizens of our country during the half century past, and

Have in the present years of world war, striven, often to the limit of endurance, to provide the means for training and implementing the united effort of our nation with other liberty loving nations to abolish overlord ideology from the earth, and

Now view the future with some misgivings born of knowledge that intense sufferings of the present are direct consequences of the past which, however, is rich in lessons that should help to avoid repetition of mistakes that have temporarily overpowered good with evil, and

Now with the courage of trained and visional minds will acknowledge the debt to our own and to other nations for training received and opportunities offered to become leaders in the advancement of living standards.

All must resolve to unite in the assertion of proven fundamental principles which created *The American Way of Life* in a nation known to the world as the *Land of Opportunity* and to promote the practice of these principles without restriction in the more intimate future relations of this country with all nations to aid in the development of *A World of Opportunity* in which man will have the right to *Life, Liberty and The Pursuit of Happiness*.

Only in such a World no overlord can gain strength sufficient to precipitate another world war, and

On behalf of the *American Society of Civil Engineers*, we hopefully offer you our full cooperation in continuing efforts to that end.



THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, by R. M. Gates, *President*: From the first paper read before your Society it has aggressively pursued and encouraged research on insulating materials, heat losses, ventilation and air conditioning for the purpose of developing standards for good practice.

The physiology as well as the physics of heating and ventilating rooms and auditoriums have received consideration.

For all that your Society has done to make life indoors safer, more comfortable and healthier, it deserves high commendation.

The *American Society of Mechanical Engineers* is pleased to congratulate the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on its 50th birthday and to prophesy an even greater success in the next half century.

A toast to your Society's Past, Present and Future!



AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS, by H. H. Henline, *National Secretary*: The *American Institute of Electrical Engineers* extends to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS its sincere congratulations upon the attainment of a record of a half century of splendid accomplishments in a large and important division of engineering.

During the last half of the 19th century engineering education and engineering practice in the United States were developed rapidly. There was a natural tendency for the engineers in each of the principal divisions of engineering to organize in order that technical developments might be encouraged.

Following the organization of the first few societies in the principal large divisions of engineering, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, organized in 1894, was one of the first societies to encourage intensive developments in a more specialized field, having as its objective the advancement of the science of heating, ventilating and air conditioning.

It deserves the congratulations of engineers and others for its outstanding contributions through the stimulation of research and invention, and the development of codes and standards in a division of engineering having a vital bearing upon the health and general welfare of our entire population.



AMERICAN INSTITUTE OF MINING AND METALLURGICAL ENGINEERS, by A. B. Parsons, *Secretary*: May I, on the occasion of the 50th anniversary of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, through you, convey to your officers and members the compliments of the *American Institute of Mining and Metallurgical Engineers*.

At this momentous hour in history the world cannot but be impressed with the fact that the achievements of engineers have been responsible, in large part, for making modern war the deadly and dreadful thing it is.

However, we know that these same achievements, in the era of peace that surely will come, can make of the world a better place in which to live. In this knowledge, engineers of all kinds can take comfort.

I am sure that your splendid Society will long continue to contribute in an important way to the sum total of human well-being and happiness.



AMERICAN MEDICAL ASSOCIATION, by Carl M. Peterson, M.D., *Secretary of Council on Industrial Health*: An occasion of this kind calls for the sincerest kind of congratulations and the hope that the same success which has marked your first half century will characterize your growth and development henceforward.

Please accept the best wishes on this commemorative occasion of the *American Medical Association*.



AMERICAN INSTITUTE OF ARCHITECTS, by E. I. Williams: I represent the President of the *American Institute of Architects*, Raymond Ashton, to bring you the greetings of the Institute. Our Societies have very much in common. When the *American Institute of Architects* was formed and organized, in 1857, the purposes set forth were the advancement of art and science. It was to be a part of the proceedings of each meeting of the Institute that a paper would be read for general discussion, and I think it is of some historical interest that at the very first meeting of the *American Institute of Architects* Richard Upjohn, the grandfather of our Hobart Upjohn, Fellow of the *American Institute of Architects*, read the first paper, and that paper was on furnaces and on heating. The discussion centered around the possible danger of using tin pipes and as to whether or not ducts should be built into the walls. That led to a considerable discussion.

The profession of architecture is very old. The profession of heating and ventilating, with all due respect to what the Romans and others have done, is relatively new in the light of standards here in America, as exemplified by that first meeting of the *American Institute of Architects*.

The future is truly unlimited. Our American way is a way of cooperation. There is free enterprise and competition but withal there is an understanding that we work together.

It is my hope, expressed for the *American Institute of Architects*, that the great future, with its unlimited possibilities of cooperation, will see greater cooperation and greater understanding between our two Societies; and, gentlemen, I bring to you not only the greetings and congratulations of the *American Institute of Architects* but also the great hope for the future that we, in serving ourselves, may also serve our Societies and our country.



THE AMERICAN PHYSICAL SOCIETY, by Karl K. Darrow, *Secretary*: It is a pleasure to bring you the congratulations of the *American Physical Society* and of the *American Institute of Physics* on this auspicious occasion. I am sure that you would like to be addressed in an appropriate way by a very great physicist; and this I have succeeded in arranging. As, however, the physicist in question is no longer living, I shall have to read to you his words. He was William Thomson, better known to this generation as Lord Kelvin; and 20 years before your Society was born, he gave the presidential address from which I am going to quote. I mention that it was given before the *Society of Telegraph Engineers*, as this will explain a metaphor which otherwise might mystify you.

"Architecture is not commonly called a branch of engineering at all. I think it unfortunate that the public do not regard architecture as a branch of engineering. When architects come to regard themselves as engineers, and when the public expect them to act as engineers, let us hope they will give us buildings not less beautiful and not less interestingly connected with monuments and traditions of beauty from bygone ages than they give us now. Then invalids too ill to walk, or ride, or drive out of doors, or to be benefited by the beautiful scenery of Mentone or Corsica or Madeira, will not be expatriated merely to avoid the evil effects of the indoor atmosphere of England. Then people in good health will not be stupefied by a few hours of an evening at home in gaslight, or of a social reunion, or by one hour of a crowded popular lecture or meeting of a learned society. Then in our hotels, and dwelling-houses, and clubs, we shall escape the negatively-refreshing influence of the all-pervading daily aerial telegraph, which prematurely transmits intelligence of distant and future dinners. The problem of giving us within doors any prescribed degree of temperature, with air as fresh and pure as the atmosphere outside the house can supply, may not be an easy problem; but it is certainly a problem to be solved when architecture becomes a branch of scientific engineering."

So spoke Lord Kelvin in 1874, and now his words sound more than a little amusing; and amusing they are, precisely because your profession exists. Your profession has fulfilled all of Lord Kelvin's dreams, and your Society, I doubt not, has been instrumental in fulfilling them. Of course his dreams did not include that of cooling houses down in hot weather, for he lived in a land where the climate is chilly and the houses chillier yet. We in this country have that dream and I am glad to see that your program shows that you are interested in fulfilling it. May you soon bring us the day when the heat of summer will be no more fearful than the cold of winter, because we shall experience each only in passing from one building to another!



AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, by J. F. Stone: It is my honor and great privilege to come before you charged with messages from the *American Society of Refrigerating Engineers*. We offer you our congratulations on a great past and our certain belief in your greater future. We have many interests in common, many members in common, and we shall go forward, we hope, with you. We know you will go forward and that you will be contributing, as one of my predecessors pointed out, to the development of this country and our way of doing things.



AMERICAN SOCIETY FOR TESTING MATERIALS, by J. R. Townsend: At the request of the President of the *American Society for Testing Materials* and on behalf of the

Executive Committee of that Society, I wish to extend to you their hearty congratulations on your 50th birthday.

It seems strange to speak to such a young and vigorous organization with such a promising future and to realize that it is fifty years old. I believed that the *American Society for Testing Materials* was an old organization but I find that we are only 42 years old. We have eight more years to go before we can reach this fifty-year milestone.

As you know, gentlemen, the *American Society for Testing Materials* is engaged in the study of materials and methods of tests and preparations of specifications. Many of you are members of the *American Society for Testing Materials*. In a sense we collect and collate the data that you people use in your work. We also devise the standard specifications for the purchase of material so that the materials will be uniform in quality and give you uniformly high performance.

The *American Society for Testing Materials* is trying to be up to date in every respect and to provide for the engineering profession and the major engineering societies data on up-to-date subjects. I am happy to report to you that the *American Society for Testing Materials* is currently studying and standardizing plastics, ceramics, and bonded plywood. A new committee has just been formed on adhesives. We have just formed another new committee on powder metallurgy for bearings, and we are now making a very active study of mildew and fungi attack. This last subject is of great interest, of course, to those who must work under tropical conditions. Everyone knows that mildew forms nearly everywhere, and is of extreme importance to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

We have also under way exposure tests in various places in the United States and in the tropics, in order to gain information concerning engineering materials.

In concluding, I would like to tell you a story which I think is quite pertinent to the message that I wish to bring to you.

A solicitor for an extension university was walking through the countryside and trying to sell to farmers an extension course in farming. As he approached one farmer he gave his usual sales talk and the farmer said, "Well, what good will this course be to me?"

The salesman said, "Well, it will make you a better farmer."

"Oh," said the farmer, "I only farm half as well as I know how to now."

So gentlemen, we of the *American Society for Testing Materials* will ply you with data and all the assistance we can, and I am sure you will do the best engineering job, because certainly the future seems very bright, and I know that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS will have a very important part.



AMERICAN GAS ASSOCIATION, by H. O. Loebell: I am extremely grateful for the privilege of representing the *American Gas Association* today, because it gives me an opportunity to express the warm admiration which I, as well as all the other members of our Association, have always had for your Society. A great deal of our enthusiasm for your group is motivated by a sense of appreciation. You have done much to advance the science of heating and air conditioning, and these over-all efforts have directly brought about added prestige and wider acceptance of gas fuel.

But apart from these contributions, and entirely divorced from any selfishness which we might have in your accomplishments, we would like to pay sincere tribute to you as a group of zealous, earnest men who, for fifty years, have steadfastly adhered to one objective—that of developing better engineering methods by which to serve the public.

We admire particularly your scholarly approach to every problem, your thoughtful, careful planning, minute attention to detail, and ingenuity, which are always in evidence in your undertakings.

These same attributes have underwritten their success. For example, it has been my privilege to read many reports on research sponsored by your Society, as well as

numerous technical papers that you men have contributed which have been widely circulated, and it seems to me that in all of them the clearly manifest motive has been a sincere desire to make a contribution that would inure to the benefit of your fellow men, and in that unity of purpose lies, we believe, your strength.

You are a voluntary organization. It is not subsidized nor is it endowed. Therefore, it must reflect the enthusiasm and purpose of the men who are members of the Society. Any association that is a voluntary association and has survived fifty years, must have been of service to the industries to which it caters and which it stimulates, and to the public who are the ultimate beneficiaries. It must have made very valuable contributions; otherwise, it would never have been maintained, particularly during the past fifty years, a period during which all kinds of economic conditions and changes have presented problems which were a test of the strength of any voluntary organization.

We of the *American Gas Association* have watched with a great deal of interest and approval the efforts made by your Society. Naturally, we always felt a close kinship with you. We have a common interest, in that our major efforts have been dedicated to the refinement and betterment of one of the basic needs of man—shelter. You, as well as we, focus your efforts on service to millions of people. In many instances our paths have merged and our interests have become identical. Scientific knowledge and engineering skill are necessary for the successful interpretation of our common problem. Thus we have borrowed a great deal of information from you and we have in turn endeavored to give you full cooperation, supplementing the information and knowledge available with our own findings and developments that had to do with the use of gas.

After the war we look forward to an even closer relationship and cooperation with your Society, because we believe that there will be accelerated activity in the major field to which our common efforts are dedicated—the home. New developments will be brought out, new appreciation of comfort and convenience will be engendered. This will be the natural aftermath of war, because war always accelerates change.

For example, we both know that a much larger percentage of homes will be automatically heated. Accelerated progress is anticipated in the field of air conditioning, a comparatively new field, wherein you have already made so many valuable contributions.

Our fuel is so readily adaptable to both heating and air conditioning, and the two services complement each other so readily, that we look forward to going on and on with you for the successful fulfilment of this year-round comfort objective.

You have pioneered the awareness of proper heating and cooling and ventilating. The difficulty prior to the war has been mainly economic; but if the plans being currently discussed are realized 100 per cent, with resultant high national income and consequent surplus spendable income for a greater percentage of the people, these developments you have already brought forth will have a wider and more fruitful acceptance.

You are now ahead with all the new scientific developments of materials and interpretation, and you will therefore be able to render still greater service to a public which has become constantly more aware of comfort standards and needs, and in the fulfilment of this objective I should like to assure you that the *American Gas Association* will give you wholehearted support and cooperation.

In closing may I say that the beacon light of progress you have established and maintained for the past 50 years is an assurance that you will acquit yourselves nobly in the postwar era, which might well be the highlight of your next fifty years.



AMERICAN STANDARDS ASSOCIATION, by Dr. P. G. Agnew, *Secretary*: I bring you greetings from the *American Standards Association* and congratulations upon rounding out half a century of service.

This service has been outstanding in the field of research and development. It has had to do with one of the oldest of the arts—the art of making warm and comfortable the air in which man lives and works. Your half century has seen this art develop into a full-blown engineering science. This has come about, in large part, as a result of the labors of your Society and its members.

Our Association, which has just celebrated its Silver Anniversary, has enjoyed close cooperation with your Society almost from the start. The two organizations are now affiliated; and are working together on many important undertakings in the development of standards. We look forward to a still more productive cooperation in the future.



DR. R. R. SAYERS, Director, Bureau of Mines, U. S. Department of the Interior, Washington, D. C.: In congratulating the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on its Golden Anniversary I wish to pay tribute to its officers and its more than 3,000 members who today are playing irreplaceable roles in helping win the war on the home front and on the battle front. In peacetime as well as wartime, your Society has had as its meritorious objective the advancement of the art and science of heating, ventilating, and air conditioning, or in other words making this world a better place in which to work and live.

America's shoulder-to-shoulder fight with the Allied Nations would not be nearly so effective were it not for the great strides made in harnessing temperature, humidity, and air movement. Our stores of blood plasma are protected, equipment can be tested accurately for extremes of heat and cold before it is sent our armed forces, food is safeguarded, the manufacture of complicated scientific instruments is facilitated. The list is endless.

Fifty years of progress lie behind, but many years of progress lie ahead. In the postwar world I see an unlimited field of research in which your Society's hallmark of quality will be imprinted in every betterment pointing to longer, happier lives for all Americans.



U. S. DEPARTMENT OF THE INTERIOR, Bureau of Mines, by ARNO C. FIELDNER, *Chief, Fuels and Explosives Service*: Greetings to the Officers and Members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on this happy occasion of the commemoration of your Golden Anniversary. It marks the passing of a half century of very practical service to the American people in providing one of the primary needs of existence, namely, the maintenance of a comfortable temperature and healthful atmospheric conditions in homes and working places, as well as in public buildings where people congregate. The work of the Society is related intimately to the everyday life of the average person who has benefited greatly from its work.

My personal contact with the Society began with the establishment of the Research Laboratory at the Central Experiment Station of the Bureau of Mines at Pittsburgh, Pa. It was my privilege to take part in behalf of the Bureau in co-operative research problems with the laboratory staff, first under the pioneering direction of Dean John R. Allen and then under the genial direction of Dean F. Paul Anderson and subsequently under the direction of F. C. Houghten, now a Lt. Commander in the Navy. The most important fruits of this cooperation were the famous comfort charts of the Society, which have had widespread recognition as showing in convenient and usable form the effect of temperature, humidity, and air movement on the comfort and physiological condition of persons working or at rest under various combinations of these factors. The results of this investigation were of special value to the mining industry because of the high temperature and high humidity prevailing in many deep mines, which called for special measures to ameliorate these trying conditions for the workers in these mines. The mining in-

dustry also owes much to one of your distinguished members, Willis H. Carrier, and his associates for developing mechanical methods of conditioning the atmospheres in very deep mines, thus making it possible to recover valuable minerals at depths where formerly it would have been impossible to conduct operations on account of the high temperature and high humidity in the working places.

The studies in the psychrometric chambers at Pittsburgh have been of wide usefulness in homes, in many industries, and in naval warfare. They illustrate the fundamental interest of the members of the Society in the technical advancement of their profession and their desire to contribute to the general welfare.

I am most happy to send these greetings and to extend my best wishes for the future.



NATIONAL RESEARCH COUNCIL, by W. F. Durand, *Chairman*, Division of Engineering and Industrial Research: It is with interest that we note the fact that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is celebrating its Golden Anniversary. The substantial accomplishments of the Society in the *Past* and the activities of the *Present* give every assurance of increasing usefulness in the *Future*.

The Division of Engineering and Industrial Research of the *National Research Council*, of which the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has long been an affiliate through representation in the membership of the Division, takes great pleasure in expressing to the Administration and membership of your Society its most cordial greetings, with an expression of its desire for continuing and increasing cooperation between our respective organizations, and of our full confidence in the continuing growth and usefulness of your Society through the coming years.



THE NATIONAL BUREAU OF STANDARDS, Department of Commerce, by R. S. Dill, Washington, D. C.: The *National Bureau of Standards* extends its greetings and good wishes for another 50 years of progress such as the Society has had in the past 50.

Insofar as the discovery and publication of the truth is concerned, the interests of the Bureau of Standards and of the Department of Commerce and those of this Society coincide. I believe that is about the story.



AIR CONDITIONING AND REFRIGERATING MACHINERY ASSOCIATION, by Mary Jane Stewart, *Secretary*: On behalf of the officers and members of the *Air Conditioning and Refrigerating Machinery Association*, may I extend greetings to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on the occasion of its 50th Anniversary.

We congratulate the Society on its fine record of achievements and send best wishes for the future.



BITUMINOUS COAL RESEARCH by J. E. Tobey, *Director*: I want to congratulate the ASHVE on its 50th anniversary. We feel in the coal industry very close to the ASHVE, and we know what a remarkable job you have done.

I want to let you in on a little secret about bituminous coal. Bituminous coal is the basic fuel of America. We are finding it out now more during the war emergency than ever before.

In a way, it is unfortunate that bituminous coal is not dangerous to handle. The great flexibility of coal in that it will burn almost anywhere, in anything, under

any conditions, has been the greatest handicap to it. That is a strange paradox, but it is true. If it were dangerous to life and limb we would have found out how to carburete it with air with great precision and burn it with high efficiency.

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS have done remarkably well in the distribution of heat and in the development of burning equipment for the more hazardous fuels, gas and oil. But in the case of solid fuels, the development of burning equipment has been much neglected.

We are all responsible for this situation. The coal industry has likewise failed to promote better equipment. Today we realize this mistake, and through *Bituminous Coal Research, Inc.* we are making every effort to correct it. It is not an easy task—there is no question about that. It is not enough to have a pretty exterior on heating equipment or to have compact and accessible devices. We must perfect the combustion side of the equipment.

So I bespeak your cooperation in that connection, and I can assure you that *Bituminous Coal Research, Inc.*, will work closely with you to the end that greatly improved heating equipment for bituminous coal will be available for all types of housing, including low cost houses.

Great strides have been made since 1923 in the development of residential stokers, and about a million of them are now in use. The Stoker Industry deserves high praise for this achievement.

In *B.C.R.'s* new Five-Year Program, which is being launched now, we have broken down the research work into twelve major groups and the number one group is domestic or residential heating. We have gone quite a way in the development of stoves. About 40 per cent of the homes in America are still heated with stoves, and not enough improvements have been made in them. In this connection, *B.C.R.* has developed a new principle of combustion for stoves which promises to be adaptable to furnaces and boilers. You will be interested in the progress that is being made because of its vital importance to you.

I shall not enumerate the other groups, but will merely say that they have to do with industrial uses, railroad locomotives, and gas turbines. In these groups we have covered the entire field of research that is needed for the better use of coal.

I might say we have gone a long way in learning how to burn coal in large furnaces—furnaces so large that you could put this entire assembly inside one of them. Here has been done a remarkable job in burning coal efficiently, but we have not done so well when it comes to burning a little bit of coal, in small hand-fired fire boxes.

Our attitude is humble; we are starting at the bottom and trying to work up, and I assure you that the bituminous coal industry wants and needs your cooperation. I wish you a most successful meeting.



COPPER AND BRASS RESEARCH ASSOCIATION, by T. E. Veltfort, *Manager*: The *Copper and Brass Research Association* and its member companies extend felicitations to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on the occasion of its Golden Anniversary.

Since the beginning of the *Copper and Brass Research Association* about 22 years ago, we have appreciated the friendly relations between our two organizations. Such cooperation should be of particular value in the coming period of reconversion to peacetime activity when both of our industries will have many difficult related problems to solve.

Best regards and wishes to you and your organization for another half century of success.



THE ENGINEERING SOCIETY OF DETROIT: To the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on its 50th Anniversary The *Engineering Society of Detroit* sends cordial greetings and hearty congratulations.

Your past is a splendid record of service and achievement; in the present your members are doing their full share toward winning the war; for the future you have a unique opportunity to contribute still more toward human health and comfort.

We are proud to have your Michigan Chapter as one of our Affiliate Groups and have appreciated its contributions to our Society and to the engineering life of this community. We salute you.



ENGINEERS' CLUB OF PHILADELPHIA, by Lee P. Hynes: I am very happy to be able to bring the greetings of the Engineers Club of Philadelphia on this occasion of your 50th anniversary. The Engineers Club is much older—some 20 years older—and has seen the birth and the growth of our great Society. I personally am only a youngster in the ASHVE, having been a member for only 21 years; but that has been long enough for me to feel, as I know you all feel, a tremendous pride in the great accomplishments of our Society.

The Engineers Club feels a pride in having the virile Philadelphia Chapter of the ASHVE as one of its affiliated society members. We also feel pride in having a Philadelphian act as President of this Society, following in the steps of other noted Philadelphians who have creditably filled that position.

At the Engineers Club we have what is known as the Philadelphia Plan, whereby fifteen local chapters of national societies are welded together in the Affiliated Council of Engineering Societies. This has brought about a cohesion among engineers which we believe is going to be increasingly important in America. It was started nearly 20 years ago and has gradually developed. At the present time we believe it is very vital to our community. We believe that it points a way in which engineers can bring their united efforts toward all matters which are of common interest to our profession, to our communities, and to our nation.

The Engineers Club looks with grave concern at the critical times which lie ahead, but also with faith that ways will be found to solve the problems that face us all. We believe that greater unity will be worked out among the great number of technical men in this country and that it will be good for our country. We have faith that the ASHVE will be found in the forefront of any such movement and will do its part splendidly in the future as it has in the past. In conclusion I want to bid you goodspeed.



ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA, Pittsburgh, Pa., by K. F. Treschow, *Secretary*: On behalf of officers and members of our Society, may I extend our heartiest congratulations on the splendid work your organization has done for the period of years it has been in existence. We know that you will continue to lead in your branch of the profession, and extend our best wishes for your future success.

I am presenting your invitation to attend your anniversary meeting to our Board of Directors at its December meeting, and if at all possible, you may rely on our having a representative present.



HEATING, PIPING AND AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION, by G. P. Nachman: It is certainly a great privilege for me to be able to bring greetings to you from the *Heating, Piping, and Air Conditioning Contractors National Association*. The members of our association have been in the very enviable position

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HEATING, PIPING AND AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION, by G. P. Nachman: It is certainly a great privilege for me to be able to bring greetings to you from the *Heating, Piping, and Air Conditioning Contractors National Association*. The members of our association have been in the very enviable position

of having assisted in the foundation of this Society and having watched and helped it grow throughout the past 50 years.

On looking over the souvenir program I realized that, unfortunately, I am not old enough to know all these men, but I have tried to find a few of the fellow members who are. I found that the cooperation of our two associations had really been just that. Of the 49 past presidents of this Society, at least nine have been members of the *National Association of Heating, Piping, and Air Conditioning Contractors*. In your list of charter members, of the 75 men listed, at least 14 were also members of the *Heating, Piping, and Air Conditioning Contractors Association*. So that this cooperation has been one which has extended from the very birth of this Society.

Fortunately, many of the men who have been successful in the contracting field have also been very outstanding engineers, and they have served this Society well, in assisting to build up the technical end. This was necessary even in the period of 50 years ago. The program and cooperation of the Research Committee of this Society has been especially helpful to those of us engaged in the contracting end. I am sure that this cooperation will continue for many years.

This year, in Cleveland, the *Heating, Piping and Air Conditioning Contractors Association* will celebrate its 55th anniversary. We extend to you a cordial welcome to come to Cleveland and help us celebrate that event.

Accept our very best wishes for the next half-century of service and research to our industry.



**ILLUMINATING ENGINEERING SOCIETY**, by D. W. Atwater: For the *Illuminating Engineering Society* to greet you at your 50th Annual Meeting is a very happy privilege. Our two organizations have much in common. We are both striving to make the world a better and a more comfortable place in which to live.

In nature light and heat are almost always intimately associated. Thus it is not surprising to find the headquarters of the *Illuminating Engineering Society* and the headquarters of your Society located in the same building in New York.

While we have much in common we also have some differences. You create heat, for example, and we, looking upon it as a necessary evil, try to eliminate it. Naturally this can and did lead to conflicting opinions and recommendations. It was aggravated by the growing appreciation and demand for lighting systems which would provide adequate illumination for efficient seeing.

Anticipating controversies which might develop between the illuminating and the heating and ventilating engineer, our two societies several years ago, formed a joint committee to frankly and honestly discuss the facts involved. The results have been most gratifying.

Our contact with your society has developed the greatest respect for your organization. In fact, we are copying some of the things which you are doing. With your guidance we are about to embark upon a program of research. Although not as elaborate as yours at least it is a start.

Again we were very much impressed with your most excellent handbook and have appointed a committee to start the preparation of a similar publication on illuminating engineering as soon as conditions permit.

It is indeed a pleasure to be here this morning and on behalf of the officers and members of the *Illuminating Engineering Society* to offer our heartiest congratulations on your Fiftieth Anniversary and to wish you continued success in the good work which you are doing.



**INSTITUTE OF BOILER AND RADIATOR MANUFACTURERS ASSOCIATION**, by R. E. Ferry: Cordial congratulations and greetings, on your fiftieth anniversary are extended by the

*Institute of Boiler and Radiator Manufacturers.* Few organizations can match the record which the Society is now celebrating, for length of time and because of the widespread recognition of your contributions to comfort and health.

The profession of heating and ventilating engineering is an old one. The problem of providing artificial warmth existed at the dawn of civilization. The history of the central heating system may be traced back over two centuries. What you have accomplished as an organization during the past 50 years is too well known to require detailed emphasis at this time, and during the last three years your contribution to the war effort has marked the Society and its members as a vital force in our country and the countries of our allies.

A larger task, however, lies ahead for you and for all of us who are in any way connected with this industry. I am referring to something greater and even more important than the imminent\* problems arising out of the conversion from a war to a peace economy, and those problems will be serious enough. A larger task in which we all have a vital part is to translate into practical terms the knowledge which has been accumulated over the past 50 years, to the end that everyone, whatever his economic level, may be able to enjoy the maximum of comfort and health which he can possibly afford.

The records of your meetings and the meetings of other organizations such as the one I represent, reflect a tremendous amount of research and technical data designed to produce the ideal indoor environment for every type of residential, commercial, and industrial building at the minimum expense. None of us can say that that end has been accomplished. The best that any of us can hope to accomplish is to add our contribution scientifically and practically to what others have done in the past.

You, as a Society, have certain objectives: first, to find facts relating to heating, ventilating, and air conditioning; and, second, to publish the facts and make them widely known. Those objectives are identical with those of the *Institute of Boiler and Radiator Manufacturers*. We also have concentrated on research projects involving steam and hot-water heating system problems in collaboration with the University of Illinois.

How can we best coordinate the research work done by each organization and by others so as to place the consumer in a position to benefit even more largely from the results which our organizations have attained? This is perhaps the time for analysis of what the most effective procedure shall be for the next year, five years, or 50 years. To be more specific, there is no single source available today where an architect, engineer, or consumer may obtain authentic, concise, and practical information as to all the details of the most efficient installation of a complete steam or hot water heating system for a building of a given type or size. In saying that I would not have anyone misunderstand me and get the impression that I am discounting in any degree the value of *THE GUIDE* which we all recognize.

The *Institute of Boiler and Radiator Manufacturers* is working now with others on the development of installation manuals to supply such information, based on the best available data derived from our research work at Urbana and all other possible sources. We feel that there should be a real interest on the part of your Society in the development and distribution of this type of practical information. The cooperation of your Society would add immeasurably to the effectiveness of this work, and we want to find the best procedure for obtaining your fullest cooperation.

The members of the Institute which I represent are active in the work of the Society, and I am sure they share with me the desire to have the work which one organization is doing supplement to the highest possible degree the work of the other. We shall count it a privilege to collaborate closely in your activities, and extend best wishes to you for another half-century of successful endeavor.



INSTITUTION OF HEATING AND VENTILATING ENGINEERS OF GREAT BRITAIN, by Richard Crittall, *President* (Cable): *Institution of Heating and Ventilating Engineers of Great Britain* send hearty congratulations and good wishes to all members of the American Society on completing 50 years active work for furthering the well being of their fellowmen.



INSULATION BOARD INSTITUTE, Chicago: RESOLUTION \*—WHEREAS, our friends in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS are celebrating the fiftieth anniversary of the founding of the Society,

BE IT THEREFORE RESOLVED, that by this resolution there is extended to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and its members the congratulations of the *Insulation Board Institute* on this fiftieth anniversary, and

BE IT FURTHER RESOLVED, that the *Insulation Board Institute* hereby expresses to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS its appreciation of the contributions made by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to the development of heating, ventilating, insulation and air conditioning arts and sciences.



NATIONAL ASSOCIATION OF BUILDING MANAGERS, by Major G. A. Shetton: On behalf of the membership of the *National Association of Building Managers*, I extend greetings to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on its Golden Anniversary. The accomplishments of the Society have been a distinguished contribution to the nation. Its achievements in the world of engineering have made the dreams and theories of yesterday the realities of today. Our industry is not unmindful of its debt to the Society and its capable membership, for the advances in design and practice which have enabled us to service vast populations comfortably within this country's great skyscrapers.

Four factors contribute immeasurably to the introduction of the skyscraper. They were: The perfecting of the steel frame, the first requisite to the multiple-story improvement. The development of this steel basket on which walls could be suspended to serve the sole purpose of protection against the elements was the basic essential to sky-piercing construction.

The development of the elevator was a second essential. Those versed in research tell us that the first elevator of all time was built by Archimedes about 200 B.C. Thus, it has taken centuries to arrive at the high speed, variable voltage, automatic equipment of today, and in between, we had the steam-operated and the hydraulic, which permitted us to install vertical transportation, without which our ceiling was decidedly limited.

Two factors of equal importance remain: electric light, and heat. Thomas A. Edison solved the former for us, and you men and those who have gone before you have solved the latter.

The splendid refinements which you have brought into the heating and ventilating field, and, as a consequence, into our skyscrapers, have been a source of great satisfaction to our tenants and to us.

Today, it might interest you to know that there is spent annually in the office building field of this nation alone, the sum of \$5 million dollars for heating and ventilating. So you see we have a huge stake in the problems which you are so ably solving.

\* Unanimously adopted by the *Insulation Board Institute* at its meeting held in New York, N. Y., January 20, 1944, signed by Stuart W. Ralph, President and Paul D. Close, Technical Secretary.

We take satisfaction in the cooperative spirit which exists between your Society and our Association, and in congratulating the Society on its 50th birthday we welcome the occasion to wish it continued great success down through the years.



NATIONAL ASSOCIATION OF FAN MANUFACTURERS, by H. E. Barth: The *National Association of Fan Manufacturers* has requested me as its representative to convey the greetings and best wishes of its members to the officers and members of this Society on the occasion of its golden anniversary. The engineering data published by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and disseminated in *THE GUIDE AND SOCIETY TRANSACTIONS*, as a result of constant research and scientific development, have been of inestimable value to the fan group, as well as to other members of the heating, ventilating, and air conditioning industry. The data published by the Society are accepted everywhere as basic and fundamental, and are used by engineers, manufacturers, contractors, and consumers alike. The activities of this Society have in no small measure contributed to the health and comfort of mankind throughout the world.

With the dawn of a new day the future represents both a challenge and a responsibility, and it is safe to predict that this Society will forge ahead on a broad front, with accomplishments even greater than those of the past.

Hearty congratulations to all of you from the *NAFM*.



NATIONAL DISTRICT HEATING ASSOCIATION, by John F. Collins, Jr. *Secretary*: For a half century this great institution, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, has by its research, its meetings, and its publications shown the way to greater comfort and better health for the human race both here and abroad. For more than a third of a century the *National District Heating Association*, whom I now represent here, has benefited from the friendly and cooperative spirit of the officers and members of the Society, a goodly number of whom have also been members of our organization.

So it is an honor and a pleasure for me to say, for the *National District Heating Association* at this golden anniversary of yours, congratulations. You have truly had a glorious past. We wish and assure you that you will have an even more glorious future.



NATIONAL WARM AIR HEATING AND AIR CONDITIONING ASSOCIATION, by H. P. Mueller, *President*: Upon this outstanding occasion, celebrating the Golden Anniversary of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the officers, members of the Board of Directors and members of the *National Warm Air Heating and Air Conditioning Association*, salute and greet you!

Your Society began to make its many worthy contributions to the War Effort beginning 50 years ago. Without that effort, surely, the satisfactory production records of our industries in the past few years would not have been possible. The health, comfort, convenience and economy factors which dominate properly engineered and installed heating and ventilating systems are as necessary in industrial as they are in commercial and domestic applications. We are happy and pleased to congratulate you for your accomplishments.

Our most sincere good wishes to each and every member of the Society.



SMOKE PREVENTION ASSOCIATION OF AMERICA, by W. E. E. Koepler: Friends of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS: I use that word *friends* deliberately because perhaps your contacts in the heating industry with the smoke inspector have not always seemed so very friendly. I am going to try to bespeak for our organizations a more friendly relationship before I finish these brief greetings.

The *Smoke Prevention Association of America*, 35 years old, presents its congratulations to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and its very best wishes for a very brilliant future following its very notable past.

After some 18 years of effort I have been able to bring about a rather cordial relationship between the *Smoke Prevention Association* and the coal industry and substantial organizations of it in which I am interested. So I might take the liberty of presenting to you also the greetings of the *National Coal Association*, until Mr. Eavenson arrives, as well as our *Bituminous Coal Institute*, the *American Mining Congress*, and various other substantial coal groups which are interested in you. We again bespeak your interest in us, because without this coal there would not be so much heat. Perhaps you have experienced the proof of that recently.

The coal industry stuck its head in the sand at the approach of the smoke inspector, and I have done what little I could in these 18 years to overcome that. Perhaps some of you have felt the same way about it. I hope that this introduction to your organization, which is one of the first, perhaps, will lead to your returning the call, and bring about a more cordial and substantial relationship throughout your next half-century.

The coal industry is now in the throes of government regulation, and it is very onerous and very burdensome to us. We are apprehensive that bureaucracy is a menace to free enterprise in this country, and we speak with some experience.

We feel that research (and we are very much interested in your research program) is the promise of freedom in this country and in our enterprises. We do not believe that bureaucracy, with all due respect to present company, can keep pace with industry if it applies research and puts on the speed in its progress that we know industry in this country is capable of putting on.

In wishing you a glorious future, in which we hope to participate, we offer you the sympathetic understanding of the *Smoke Prevention Association of America, Inc.*, including the smoke inspectors of the United States and Canada, with an occasional kick in the pants to keep things moving and bring about that necessity which is the mother of invention and research. Our congratulations.



SOCIETY FOR THE PROMOTION OF ENGINEERING EDUCATION, by R. L. Sackett: It is my pleasure to represent the President of the *Society for the Promotion of Engineering Education*, who is President of the Carnegie Institute of Technology, and who was unable to be present because of the flu. This Society, known as the *SPEE*, is composed of something over 3,000 mostly teachers of engineering subjects, with an additional sprinkling of representatives of industry who are concerned with education particularly in the engineering field.

It is important that in any group of men in the field which concerns you there should be those who are naturally educators, those who are investigators, and those who by experience have had contact with the applications of instruments, devices, and so forth.

The engineering colleges have assisted in making contributions to the study of heat flow, a very complicated subject, as you well know. They have contributed to the instrumentation, the accuracy with which heat flow, and insulating qualities, have been measured. Your Society in this field of insulation has made a very material contribution over the years.

I was interested in going over the exhibit which you have, in observing the evolution of pamphlets and textbooks and of devices such as control valves and so forth. Anyone here who is 50 years old or over, and who knows the little red schoolhouse, can trace for himself the extraordinary evolution up to the present time to which your Society has contributed so greatly. I think there is no mark of our civilization more obvious to all of us than this contribution to comfort.

I think there might be a speech on the relationship between being comfortable and spiritual development. The man who is hungry is not very much concerned about his spiritual life until he has been fed; but I leave that only as a suggestion.

Your members have helped to raise the standards of instruction in our engineering colleges and by investigating the influences affecting them, have improved the practical application of heating devices of all kinds. One might remark that you had probably been responsible for producing more heat than any other society in the United States, but you have also been responsible for the illumination on the subject, which are both, of course, highly commendable.

You have been concerned with the physiological effects of air conditioning, the studies of fatigue, the relationship of sanitary conditions, including air and heating, to the effectiveness of men. You have produced the evidence, in these last 50 years, of your ability to contribute to this thing we call advancing civilization. You have met during these immediate years great difficulties. They have, I assume, disciplined you in meeting adversity, and you are prepared to meet those difficulties of the reconstruction period which again will contribute to your further advancement and to the high place you occupy among the engineering societies.



STEEL HEATING BOILER INSTITUTE, J. E. Axeman: I have been asked to express the sentiments of the *Steel Heating Boiler Institute* on this occasion, your fiftieth anniversary. It is therefore a deep honor and a sincere privilege for me to extend our best wishes and congratulations to you, with the trust that your accomplishments in the next fifty years will equal those of the past.



STOKER MANUFACTURERS ASSOCIATION, by E. C. Webb: I am very glad to convey to you greetings from the *Stoker Manufacturers Association* to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on this, its 50th anniversary. In comparison with the long life of the Society, ours is a young industry. In fact, it was not until about 1930 that stokers began to take an important place in the heating field. However, demands for more fully automatic and more efficient heating equipment have caused our industry to grow very rapidly, and today there are over a million units in operation, representing an investment of approximately a third of a billion dollars.

The Guide and other publications and the work of this Society have been of immeasurable value to our industry, particularly in connection with the application of our equipment, and it is certainly a real pleasure to extend congratulations to the officers and members of the Society on this historic occasion.



KIWANIS INTERNATIONAL, Chicago: The success of America's war effort depends to a great extent on the strength of its home front!

For 50 years the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has contributed mightily to the creation of a strong home front by drawing plans and speci-

fications and devising equipment intended to keep houses and buildings warm, insulated and healthful with conditioned air.

When the United States declared war against the Axis nations, the Society immediately turned its attention to the production of armaments.

Kiwanis International, with its 2,200 clubs and 125,000 members, joins the parade of organizations extending congratulations and good wishes to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on its Golden Anniversary.

Since the establishment of your society in 1894, your members have blazed a historic trail that is recognized the world over. From the days of imperfectly heated and ventilated homes and buildings, your organization has designed equipment that has brought comfort, joy and happiness to every household and office.

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has enjoyed a glorious past, but its future offers possibilities that are far-reaching. When the United Nations have successfully concluded the war, America immediately will begin thinking in terms of peace-time goods and reconversion will find home and building construction increasing by leaps and bounds.

The Society will not miss this opportunity to continue its great work of the past and to further develop and improve the technical equipment so vital to every American.

Kiwanis International maintains that America's future is secure. Its best wishes go to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in war and in peace.



THE NATIONAL RADIATOR CO., Johnstown, Pa., by Robert S. Waters, *President*: The National Radiator Co. extends congratulations to the Society, its officers and members on this occasion, the observance of our mutual 50th Anniversaries. Research work and improvement of the standards of the heating and air conditioning profession during the past 50 years are creditable achievements of the Society.

We know that while the Society convenes to consider the current engineering problems of wartime activities, it also is giving consideration to a postwar program which will establish improvements in the standards of heating equipment design, installation, and operating results.

In this aim, The National Radiator Co., pledges its cooperation during the second half-century, to design and produce improved equipment for the benefit of the consumer.



THE WISCONSIN CHAPTER of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, by F. W. Goldsmith, *President*: Birthday Greetings and Best Wishes to the Society on the great occasion of the 50th Anniversary.

May the next 50 years see an equal measure of ASHVE accomplishments and progress.

## FIFTY YEARS IN HEATING AND VENTILATING

By SAMUEL R. LEWIS,\* CHICAGO, ILL.

**B**ASED on my memory and on considerable reference to old text books and catalogues I found that eastern and central western practices in heating and ventilating 50 years ago had many differences.

The central west seems to have preferred single-pipe steam heating, while the east used two-pipe systems; usually with the return mains sealed below the water level in the boiler. Chicago had miles of apartment buildings all with single-pipe steam heating systems; most of them with brick set steel fire-box boilers.

New York had many similar buildings, but the radiators usually had two connections and two valves, and the boilers were likely to be of sectional cast-iron type. Perhaps the early two-pipe vapor systems were developed because of the difficulties in controlling steam radiators which had two valves.

The design of cast-iron sectional boilers as of 1894 shows that heat transfer surface was sought at the expense of adequate combustion chamber height. They made the boiler so full of water passages that there was not enough space left for the burning coal.

I remember being called on to correct the heating of a large Chicago residence about 1910. The sectional boiler had water-cooled heat absorbing surfaces only 14 in. above the top of the 60-in. long grate. They could not burn enough coal to heat the house. I held the boiler up on jackscrews while I wrecked the cast-iron base and lowered the grates to give a 30-in. clearance between them and the top of the combustion chamber. Refractory lined masonry was used to form a new firebox and ashpit. The results were excellent in every way, and the old boiler serves the old house with ease and satisfaction to this day.

I remember designing the heating of two large high schools within 50 miles of New York City, about 35 years ago. The contractors seemed to know nothing about steel firebox type heating boilers such as I was accustomed to use in the west. I also remember seeing in the public schools in Elmira, New York, sectional cast-iron boilers made by H. B. Smith Co., operating at 50 lb per sq in. pressure and furnishing steam to run the fan engines.

One large manufacturer made a very radical change in the design of cast-iron sectional boilers about 1905, cutting out most of the interior water passages and gaining combustion space at the expense of heat absorbing surface. These boilers had hot breechings and the design very quickly was modified to one in which there was a more logical balance between the two fundamental requirements.

All of the early sectional cast-iron boilers were short on steam liberating area at the water line and on interior water circulation from section to section.

\* Consulting Engineer. Member of A.S.H.V.E.

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1944.

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## FIFTY YEARS IN HEATING AND VENTILATING

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It is only within very recent years that the large top connecting nipple, partly below the water line was developed. With the older boilers priming usually manifested itself and much washing out of every new boiler was required. The increased steam liberating area and the coherent water circulation when the top nipples are partly below the water line have improved this matter.

Old catalogues show cast-iron sectional boilers encased in brickwork, and many round boilers equipped with steel jackets. Then steel jackets on cast-iron boilers seem to have been forgotten during many years, only to reappear in comparatively recent times with deluxe baked enamel finish.

Around the first of the century there was some question in designers' minds as to justification for use of the comparatively new vacuum system of steam heating, on which a royalty must be paid on account of patents. The Paul scheme, started in Chicago, I think, involved piping up the air vents of a single-pipe radiator system, leading to a steam jet exhauster in the boiler room for maintaining sub-atmospheric pressure when desired throughout the system. It improved many an otherwise sluggish and noisy steam heating plant and hundreds of the old systems still are in use, usually with improved exhausters using electric power.

Some one probably in the east, perhaps Warren Webster, then developed a trap for the return connection from each radiator on a two-pipe heating system so that the Paul separate air pipe would not be needed. This trap at first was of float type and its use was popularly believed to require that a royalty must be paid to the owner of the Paul patent. In those days there was much litigation concerning patents on sub-atmospheric steam heating.

The float trap had definite limitations and soon was succeeded by the vapor filled thermostatic disk and bellows type of vacuum trap which persists to this day. It is believed C. A. Dunham pioneered in the thermostatic trap.

I have a data book which was published in 1895. It carries advertisements of the principal manufacturers of heating and ventilating equipment. It gives the amount of direct radiation alleged to be necessary at various rates per square foot of glass surface, or per square foot of wall surface or per cubic foot of contents of a room. It is careful not to suggest the most approved rate, but one example cites 1 sq ft of radiation for 37.5 cu ft contents.

I know that in those days no one did much Btu work, but generally used the formula ascribed to Mills for steam, which was:

1 sq ft of radiation for each 2 sq ft of glass in outside walls, plus 1 sq ft of radiation for each 20 sq ft of exposed wall (deducting the square feet of glass) plus 1 sq ft of radiation for each 200 cu ft in the room.

For ordinary thermally circulated hot water radiation the factors changed to 2, 10 and 60 respectively. Like many old housewife medicines, this rule of thumb checks very well with the average of more scientific formulas.

Some of the Bundy radiators of 1894 were arranged for cast-iron or marble flat top plates. These radiators were made of cored cast-iron tubes screwed into cast-iron bases. They are rarely seen these days. All sectional radiators definitely had to be ordered as for steam, with bottom connecting nipples only, or for water, with both top and bottom nipples. It was believed then that a water radiator would not work if used with steam; the contrary fact not having been discovered until passage of many years after 1894.

A good deal of emphasis seems to have been given to flue type radiators, some having vertical solid flanges, while others had curved flanges intended to

encourage reception of air by the radiator in a vertical up-going stream with delivery from the face of the radiator in a horizontal direction.

One rarely sees a dining room radiator having shelves for warming the dinner plates and coffee cups behind cast-iron doors, although they were popular in the nineties.

There were long arguments about how to measure the radiating surface of an ornamental cast-iron radiator and rival manufacturers questioned the methods of measurement by their competitors. As far as I can learn it was not until well along in the present century that the obvious fact occurred to any one that each 144 sq in. of surface in a radiator gives off heat at widely varying rates depending on a great many factors. End sections being exposed to free air circulation condense more steam than interior sections. The upper areas of tall radiators never see any air but that which already has been warmed by surfaces lower down. Wide radiators heat much less air per unit of surface than do narrow ones. Steam in one large tube heats less air than when in several small tubes in the same space.

The radiator manufacturers of 50 years ago seem to have been in some agreement that a radiator might be a beautiful thing if sufficiently ornamental. One can be thankful that we have emerged from that Victorian era to the present plain, thin tube, low height type of radiator.

I cannot feel that with the present enclosed thermally circulated convector we are making progress. For efficiency in transferring dust from the floor to the wall, and for effective laying down on the job when the heating medium within it is only warm and not hot, the finned tube convector of around 1940 is unsurpassed. I vote to consign it to the limbo of forgotten things along with such names as I can cull from the old catalogue as follows:

Volunteer, All Right, Commonwealth, Hecla, Mascot, OK, Hub, Economy, Novelty, Sunray, Cataract, Faultless, Royal, Electric, Tropic, Perfect, Modern, Florida, Advance, Little Giant, Imperial, Champion, Joy, Crescent, Climax, Elite, Cyclonic.

Steam convectors in 1894, where used with mechanically circulated air, almost universally were made of 1-in. pipe screwed into cast-iron headers or bases. There was talk about that time of a new cored cast-iron prime surface convector which was to be called Vento.

All of the original convectors were difficult to vent, especially when applied to vacuum systems. Frequently the older cast-iron extended surface convectors which had been used for years with thermally circulating air, were installed in fan systems. These are no longer made except under protest. They had round pins and thin flanges and carried such names as Perfection and Excelsior and Gold.

The pipe coil fan system convectors were built in many types. One had pairs of pipe from a double chambered horizontal box base connected at the top with return bends. Another used two cast-iron headers, one as a base, for condensate and one at one side, vertically disposed, as steam supply. The Chicago schools used a special base for many years in which the steam supply came vertically through the bottom of the single chambered base casting.

Any old time steam fitter will remember the fun he had when converting some of these old pipe coil convectors originally installed for pressure service, to drain and to vent under more modern vacuum system conditions.

When Vento blast radiation came along early in the century, some research and experimenting on the jobs, in which I had no small part, had to be

encountered before the correct air venting technique was developed. Cast-iron convectors, such as Vento, do not freeze so quickly as do the more flashy thin tube, wide flanged, present day convectors, but the great weight of cast-iron gradually may cause its retirement. One may predict much future trouble with the present day convectors, made of thin steel substituted by emergency for the everlasting copper of prewar days.

Fans in 1894 were of the eight-blade, centrifugal or radial flow type, with straight blades. In selecting one of these devices from the manufacturer's rating, there was always a discount for optimism of from 40 to 60 per cent. The modern, narrow multiblade fan wheel had not been imported (I think it came from Ireland).

There had been little research on fan efficiencies and all radial flow fans had small outlets and chaotic delivery velocities at various spots across these outlets. If perchance a fan after installation, proved too small, additional steel plates could be bolted to the outer edges of the paddles, thus increasing the diameter and the noise and the power. Not much was said in catalogues about the volume and power at various resistances.

There were axial flow fans of flat disk type and of propeller type, but it was known that they were not of much use in heating systems because of failure to overcome the resistance due to heaters and long ducts.

There was little knowledge yet developed concerning refinement of centrifugal fan inlet rings, various curves in blades, etc. Research along these lines still continues, and if one can believe the advertisements there will emerge for general use, after the war, great improvement in axial-flow pressure type fans.

Warm air heating by direct coal-fired cast-iron furnaces was well developed in 1894, especially for school buildings. While thermally circulated air was used for the smaller schools, many larger school buildings were heated by mechanically circulated air.

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By FERRY C. HOUGHTEN\* (D.Sc.), WASHINGTON, D. C.

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During and following the first World War, other conditions of the atmosphere were being considered. One of the first attempts to catalogue and understand the importance of the several physical and chemical properties of man's atmospheric environment was the development and presentation (2)† to the Society of the synthetic air chart by Dr. E. Vernon Hill. This work served several useful purposes in the early development of comfort air conditioning: first, by cataloguing the several factors concerning the atmosphere known to affect the comfort and well-being of man, wide publicity and acceptance was given to their importance; second, by attempting to evaluate these factors in terms of the knowledge then available, attention was centered on the lack of any semblance of accepted quantitative data on many phases of the subject. A good example of the lack of acceptance of standardized values concerning comfort air conditioning was the wide difference of opinion concerning the effect on a person's feeling of warmth of such physical conditions of the atmosphere as its dry-bulb temperature, moisture content, and motion. Dry-bulb alone still received general acceptance. The chart went to the other extreme of accepting the wet-bulb temperature as the controlling factor. Contrasted with the views of Dr. E. Vernon Hill, as expressed in the synthetic air chart, Dr. W. H. Carrier and others, based upon theory and broad experiences gained through application in the field, held that both dry-bulb and wet-bulb were effective in determining a person's feeling of warmth. The wholesome controversies concerning these points drew attention to the then current work of Dr. Leonard Hill and others. Probably more than any other single influence, these controversies served to direct attention to the need for laboratory research in this branch of the engineering profession and industry, and eventually to the organization by the Society of its Research Laboratory in 1919.

Dr. C.-E. A. Winslow and his associates on the New York State Commission on Ventilation, and the reports of their work, served a most useful

\* Lt. Comdr., USNR, Research, Air Conditioning Section, Bureau of Ships. Also Research Division, Bureau of Medicine and Surgery. Member of A.S.H.V.E.

† Numerals in parentheses refer to Bibliography.

The opinions and assertions contained in this paper are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.

Presented at the 50th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1944.

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## FIFTY YEARS IN HEATING AND VENTILATING

By SAMUEL R. LEWIS,\* CHICAGO, ILL.

**B**ASED on my memory and on considerable reference to old text books and catalogues I found that eastern and central western practices in heating and ventilating 50 years ago had many differences.

The central west seems to have preferred single-pipe steam heating, while the east used two-pipe systems; usually with the return mains sealed below the water level in the boiler. Chicago had miles of apartment buildings all with single-pipe steam heating systems; most of them with brick set steel fire-box boilers.

New York had many similar buildings, but the radiators usually had two connections and two valves, and the boilers were likely to be of sectional cast-iron type. Perhaps the early two-pipe vapor systems were developed because of the difficulties in controlling steam radiators which had two valves.

The design of cast-iron sectional boilers as of 1894 shows that heat transfer surface was sought at the expense of adequate combustion chamber height. They made the boiler so full of water passages that there was not enough space left for the burning coal.

I remember being called on to correct the heating of a large Chicago residence about 1910. The sectional boiler had water-cooled heat absorbing surfaces only 14 in. above the top of the 60-in. long grate. They could not burn enough coal to heat the house. I held the boiler up on jackscrews while I wrecked the cast-iron base and lowered the grates to give a 30-in. clearance between them and the top of the combustion chamber. Refractory lined masonry was used to form a new firebox and ashpit. The results were excellent in every way, and the old boiler serves the old house with ease and satisfaction to this day.

I remember designing the heating of two large high schools within 50 miles of New York City, about 35 years ago. The contractors seemed to know nothing about steel firebox type heating boilers such as I was accustomed to use in the west. I also remember seeing in the public schools in Elmira, New York, sectional cast-iron boilers made by H. B. Smith Co., operating at 50 lb per sq in. pressure and furnishing steam to run the fan engines.

One large manufacturer made a very radical change in the design of cast-iron sectional boilers about 1905, cutting out most of the interior water passages and gaining combustion space at the expense of heat absorbing surface. These boilers had hot breechings and the design very quickly was modified to one in which there was a more logical balance between the two fundamental requirements.

All of the early sectional cast-iron boilers were short on steam liberating area at the water line and on interior water circulation from section to section.

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It is only within very recent years that the large top connecting nipple, partly below the water line was developed. With the older boilers priming usually manifested itself and much washing out of every new boiler was required. The increased steam liberating area and the coherent water circulation when the top nipples are partly below the water line have improved this matter.

Old catalogues show cast-iron sectional boilers encased in brickwork, and many round boilers equipped with steel jackets. Then steel jackets on cast-iron boilers seem to have been forgotten during many years, only to reappear in comparatively recent times with deluxe baked enamel finish.

Around the first of the century there was some question in designers' minds as to justification for use of the comparatively new vacuum system of steam heating, on which a royalty must be paid on account of patents. The Paul scheme, started in Chicago, I think, involved piping up the air vents of a single-pipe radiator system, leading to a steam jet exhauster in the boiler room for maintaining sub-atmospheric pressure when desired throughout the system. It improved many an otherwise sluggish and noisy steam heating plant and hundreds of the old systems still are in use, usually with improved exhausters using electric power.

Some one probably in the east, perhaps Warren Webster, then developed a trap for the return connection from each radiator on a two-pipe heating system so that the Paul separate air pipe would not be needed. This trap at first was of float type and its use was popularly believed to require that a royalty must be paid to the owner of the Paul patent. In those days there was much litigation concerning patents on sub-atmospheric steam heating.

The float trap had definite limitations and soon was succeeded by the vapor filled thermostatic disk and bellows type of vacuum trap which persists to this day. It is believed C. A. Dunham pioneered in the thermostatic trap.

I have a data book which was published in 1895. It carries advertisements of the principal manufacturers of heating and ventilating equipment. It gives the amount of direct radiation alleged to be necessary at various rates per square foot of glass surface, or per square foot of wall surface or per cubic foot of contents of a room. It is careful not to suggest the most approved rate, but one example cites 1 sq ft of radiation for 37.5 cu ft contents.

I know that in those days no one did much Btu work, but generally used the formula ascribed to Mills for steam, which was:

1 sq ft of radiation for each 2 sq ft of glass in outside walls, plus 1 sq ft of radiation for each 20 sq ft of exposed wall (deducting the square feet of glass) plus 1 sq ft of radiation for each 200 cu ft in the room.

For ordinary thermally circulated hot water radiation the factors changed to 2, 10 and 60 respectively. Like many old housewife medicines, this rule of thumb checks very well with the average of more scientific formulas.

Some of the Bundy radiators of 1894 were arranged for cast-iron or marble flat top plates. These radiators were made of cored cast-iron tubes screwed into cast-iron bases. They are rarely seen these days. All sectional radiators definitely had to be ordered as for steam, with bottom connecting nipples only, or for water, with both top and bottom nipples. It was believed then that a water radiator would not work if used with steam; the contrary fact not having been discovered until passage of many years after 1894.

A good deal of emphasis seems to have been given to flue type radiators, some having vertical solid flanges, while others had curved flanges intended to

encourage reception of air by the radiator in a vertical up-going stream with delivery from the face of the radiator in a horizontal direction.

One rarely sees a dining room radiator having shelves for warming the dinner plates and coffee cups behind cast-iron doors, although they were popular in the nineties.

There were long arguments about how to measure the radiating surface of an ornamental cast-iron radiator and rival manufacturers questioned the methods of measurement by their competitors. As far as I can learn it was not until well along in the present century that the obvious fact occurred to any one that each 144 sq in. of surface in a radiator gives off heat at widely varying rates depending on a great many factors. End sections being exposed to free air circulation condense more steam than interior sections. The upper areas of tall radiators never see any air but that which already has been warmed by surfaces lower down. Wide radiators heat much less air per unit of surface than do narrow ones. Steam in one large tube heats less air than when in several small tubes in the same space.

The radiator manufacturers of 50 years ago seem to have been in some agreement that a radiator might be a beautiful thing if sufficiently ornamental. One can be thankful that we have emerged from that Victorian era to the present plain, thin tube, low height type of radiator.

I cannot feel that with the present enclosed thermally circulated convector we are making progress. For efficiency in transferring dust from the floor to the wall, and for effective laying down on the job when the heating medium within it is only warm and not hot, the finned tube convector of around 1940 is unsurpassed. I vote to consign it to the limbo of forgotten things along with such names as I can cull from the old catalogue as follows:

Volunteer, All Right, Commonwealth, Hecla, Mascot, OK, Hub, Economy, Novelty, Sunray, Cataract, Faultless, Royal, Electric, Tropic, Perfect, Modern, Florida, Advance, Little Giant, Imperial, Champion, Joy, Crescent, Climax, Elite, Cyclonic.

Steam convectors in 1894, where used with mechanically circulated air, almost universally were made of 1-in. pipe screwed into cast-iron headers or bases. There was talk about that time of a new cored cast-iron prime surface convector which was to be called Vento.

All of the original convectors were difficult to vent, especially when applied to vacuum systems. Frequently the older cast-iron extended surface convectors which had been used for years with thermally circulating air, were installed in fan systems. These are no longer made except under protest. They had round pins and thin flanges and carried such names as Perfection and Excelsior and Gold.

The pipe coil fan system convectors were built in many types. One had pairs of pipe from a double chambered horizontal box base connected at the top with return bends. Another used two cast-iron headers, one as a base, for condensate and one at one side, vertically disposed, as steam supply. The Chicago schools used a special base for many years in which the steam supply came vertically through the bottom of the single chambered base casting.

Any old time steam fitter will remember the fun he had when converting some of these old pipe coil convectors originally installed for pressure service, to drain and to vent under more modern vacuum system conditions.

When Vento blast radiation came along early in the century, some research and experimenting on the jobs, in which I had no small part, had to be

encountered before the correct air venting technique was developed. Cast-iron convectors, such as Vento, do not freeze so quickly as do the more flashy thin tube, wide flanged, present day convectors, but the great weight of cast-iron gradually may cause its retirement. One may predict much future trouble with the present day convectors, made of thin steel substituted by emergency for the everlasting copper of prewar days.

Fans in 1894 were of the eight-blade, centrifugal or radial flow type, with straight blades. In selecting one of these devices from the manufacturer's rating, there was always a discount for optimism of from 40 to 60 per cent. The modern, narrow multiblade fan wheel had not been imported (I think it came from Ireland).

There had been little research on fan efficiencies and all radial flow fans had small outlets and chaotic delivery velocities at various spots across these outlets. If perchance a fan after installation, proved too small, additional steel plates could be bolted to the outer edges of the paddles, thus increasing the diameter and the noise and the power. Not much was said in catalogues about the volume and power at various resistances.

There were axial flow fans of flat disk type and of propeller type, but it was known that they were not of much use in heating systems because of failure to overcome the resistance due to heaters and long ducts.

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By FERRY C. HOUGHTEN\* (D.Sc.), WASHINGTON, D. C.

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purpose in that the discussion (7) of the report calls attention to the importance of the condition of the atmosphere in the enclosure rather than whether ventilation was supplied by open window or by mechanical means.

*The Comfort Chart.* It may now be said that the controversies of the late teens and early twenties, which at times became acrimonious, served in no

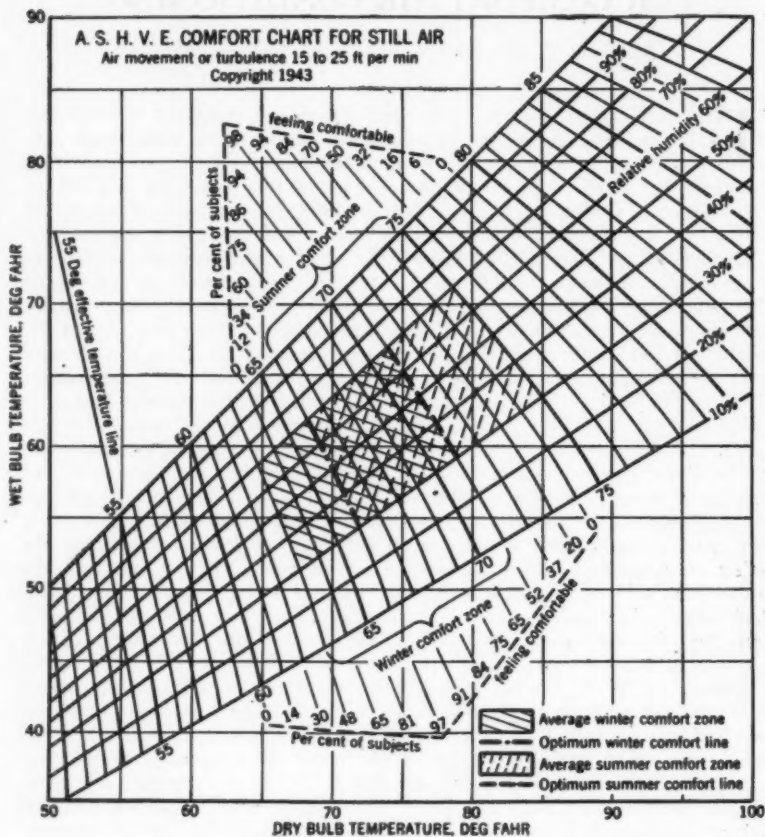


FIG. 1. A.S.H.V.E. COMFORT CHART FOR STILL AIR

small measure to supply the incentive and determination for the following twenty years of prolific research which has marked the achievement of a half century of success by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Important in the program outlined by the Committee on Research, upon the organization of its new laboratory, was the development of standards of atmospheric conditions for the comfort of man. The first of these standards to be considered was the relative effect on a person's feeling of warmth, of

the dry-bulb temperature, the moisture content, the movement or velocity of the air, and the radiant heat. This study resulted in a number of reports and the development of the well-known Comfort Chart of the Society, Figs. 1 and 2. The first report (6) of this work was presented to the Society at its 1923 annual meeting in Washington, and gave the basic chart for men in still air, stripped to the waist, as in hot industry. This was followed by additional reports (11) showing the effect of air velocity on the feeling of warmth of

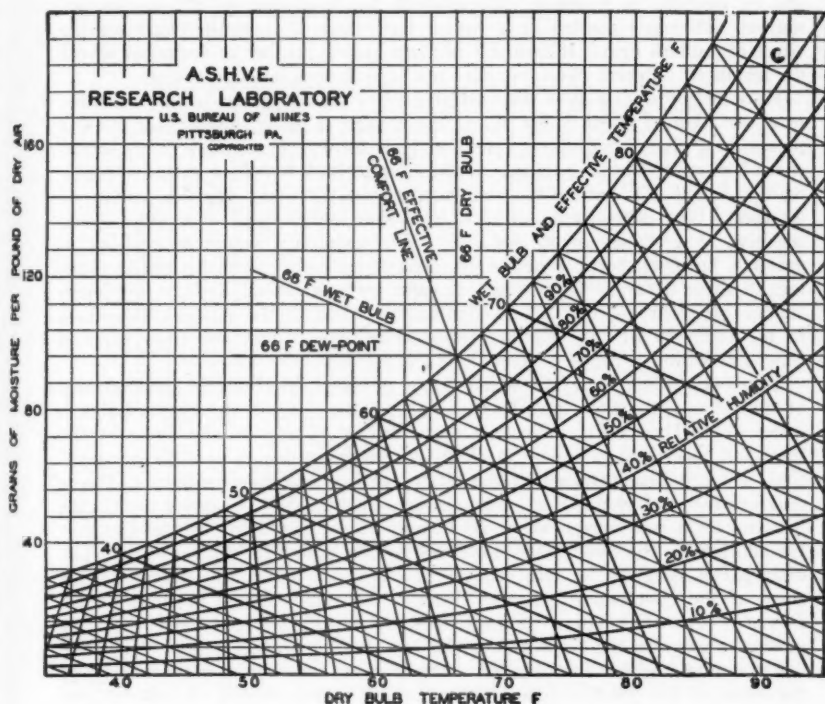


FIG. 2. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

a person in atmospheres of different temperatures and humidities. In these early studies the effect of radiant heat was made constant by insulating the walls, so that the surfaces in view of occupants were at approximately the same temperature as the air dry-bulb temperature. Later work gave corrections to the basic chart for persons normally clothed and seated at rest, so that the chart, Fig. 1, in the form in which it is now used became available to the Society in 1925 and it has not been revised since.

Continued work on the effective temperature index during the past eighteen years, both in the Laboratory and elsewhere, has shown that the slope of the lines vary to a small extent with a number of variables, including the individual

concerned at the time the observation is made, his clothing, his perspiration rate, and his activity. Other physical factors which affect the accuracy of the lines as drawn on the chart, are the air velocity and radiant heat. Data concerning the effect of the former are available, and the location and the slope of the lines may be corrected for the effect of air velocity if the velocity is known. The relative effect of radiant heat on a person's feeling of warmth, as compared with the effect of air temperature, moisture content and velocity, has been studied in recent years at the Society's Laboratory (59 and 62a) and by Bedford in England. While much has been learned, the results are not yet sufficiently conclusive to permit correcting the comfort chart for the effect of radiant heat. It probably accounts for a number of instances where the results of different observers have not been in complete agreement.

More recent studies have demonstrated (59) that a person is only sensitive to a variation of about 3 deg ET if the change is made slowly; that is, if a person in a given atmospheric condition feels ideally comfortable in his sensation of warmth, the temperature would have to be lowered or raised approximately 3 deg ET before his sensation would positively register cooler or warmer. If a person goes quickly from one environment to another he is sensitive to a change of  $\frac{1}{2}$  deg ET. This sensitivity cannot be realized in practice, however, because of the personal and physical variables cited. Taking all these different factors into account on a statistical basis, a minimum change of about  $1\frac{1}{2}$  deg ET is significant. Most of the possible variations in the accuracy of the effective temperature lines, suggested from time to time, fall within a range of from  $\frac{1}{2}$  to 3 deg ET.

Two outstanding suggestions indicating possible variations in the accuracy of the slope of the lines have been made. Studies (50) at the Pierce Laboratories, Yale University, indicate a more critical change in the slope of the lines for conditions resulting in change from insensible to sensible perspiration. This is demonstrated by the difference between the two curves, *Pierce Evaporation* and *A.S.H.V.E. Evaporation*, in Fig. 3. The main difference between these two curves, above 70 F, may be accounted for by the fact that the subjects in the two studies were not equally active or clothed alike. The discrepancy between the two curves, which would indicate a change in the slope of the effective temperature lines, is the rapid change in slope of the *Pierce Evaporation* curve at approximately 86 F, as contrasted with the gradual change in the slope of the *A.S.H.V.E. Evaporation* curve. However, the magnitude which this question may involve concerning the accuracy of the slope of the effective temperature lines, as shown in Fig. 3, will be considerably less than the three degrees sensitivity of any individual.

More recently there was a question raised concerning the accuracy of the slope of the effective temperature lines for very hot environments in the neighborhood of 95 deg ET. A check (64) of the lines in this region of the chart, both by primary sense reactions and by physiological reaction, indicates that their slope may require a slight change, possibly to the extent that 95 deg ET at 30 per cent relative humidity should be corrected to 93 deg ET or a 2 deg change. These critical studies of the accuracy of the slope of the effective temperature lines in different regions of the chart have served to demonstrate the essential accuracy of the chart as now used.

Following the original presentation of the effective temperature chart to the Society by its Research Laboratory, it is natural that there should have been a slow acceptance of its usefulness. Its gradual but increasing rate of

acceptance throughout the past eighteen years may be taken as a measure of the increasing recognition of the need for control of the atmospheric environment for human comfort and well-being. Following the original publication of the basic chart in 1923 it received attention from, and its accuracy was commented upon by those responsible for maintaining working environments in hot spaces on Naval vessels. This was demonstrated by a recent discovery in the files of the Air Conditioning Section of the Bureau of Ships, U. S. Navy, of the original paper published in 1923, together with comments indicating its essential accuracy when applied to high temperature.

*The Comfort Zone.* Prior to the development and acceptance of the Society's comfort chart, a *comfort zone* was defined as that range of temperatures

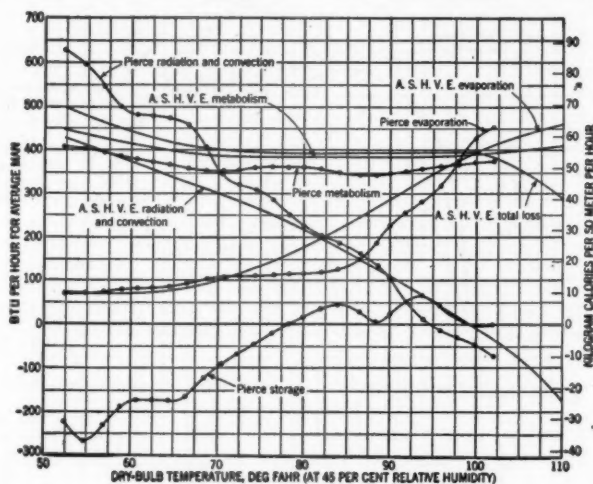


FIG. 3. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

over which most people were comfortable. This applied essentially to comfort in winter heated space. It is, therefore, natural that the establishment of the effective temperature index should have been immediately followed by a study to determine the effective temperature range under which most people are ideally comfortable. The comfort zone and the comfort line were established (9 and 15) for winter heating by the Society's Laboratory in 1923. The maximum percentage of persons was found to be comfortable at about 66 deg ET, with a decreased percentage of comfort for higher and lower effective temperatures.

With the growing application of summer cooling and air conditioning in the late twenties and early thirties, some confusion and controversy developed concerning the conditions to be maintained. The early attempts were to use the same standards for summer comfort as were used for winter comfort. This was obviously doomed to failure and the whole effective temperature idea seems to have been temporarily discarded by many in the application

of summer cooling. Two reasons may be cited as accounting for this misunderstanding: first, summer cooling requires a higher effective temperature for comfort because of the effect of acclimatization; second, a person entering a cool space from the hot outside, in attempting to be in equilibrium with the hot outside, experiences a shock because he is wet with perspiration and therefore he is not in equilibrium with the environmental air of the cooled space. The lack of understanding and knowledge of these points resulted in various practices, including the maintaining of a 57 F dew-point, and various

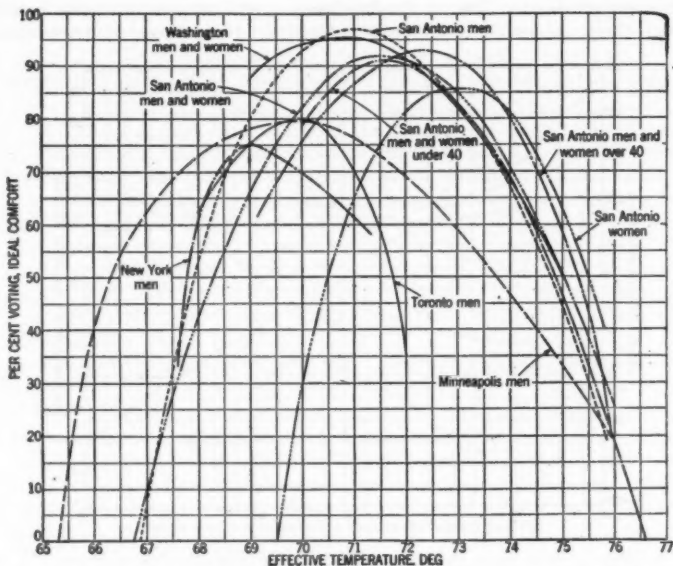


FIG. 4. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OBSERVATIONS INDICATING COMFORT

dry-bulb temperatures, based upon the weather dry-bulb temperature. By 1934 this confusion had resulted in a request that the laboratory take up a study of the requirements for summer cooling. In all, eleven papers (38, 40, 44 to 48, 53, 54, 57, and 60) were presented to the Society, dealing with the optimum effective temperature for summer cooling and air conditioning and resulting from both laboratory and field studies.

These papers made some evaluation of the effect of the cold shock experienced by a person upon entering a cooled and air conditioned space from the hot outside. They also served to show that the effective temperature lines applied for summer cooling, and that persons are comfortable in summer cooled space at effective temperatures ranging from 69 or 70 deg ET up to about 72 or 73 deg ET. Studies in a number of geographical locations indicate that while there is some variation in the optimum for different geographical regions in the United States, this variation is not great. There is some varia-

tion in the optimum effective temperature, depending on acclimatization and the age and sex of the person, as indicated by the different studies, including a survey (63) of the different studies by the Technical Advisory Committee under the chairmanship of Thomas Chester. However, the maximum variation in the optimum is not great. Fig. 4 gives optimum curves for a number of groups and geographical regions.

*Acclimatization.* In all of the studies dealing with man's physiological and psychological responses to his atmospheric environment, acclimatization has been found to be an important factor. Thus, acclimatization is shown to account for the difference in the indoor air conditioning requirement between winter and summer of from about 66 to 71 deg ET. One reason for difficulty in establishing the winter and summer comfort lines with precision is the fact that the time spent in the conditioned spaces itself tends to acclimatize an occupant. Obviously, if a person should spend all of his time in atmospheric environments conditioned to suit his comfort in either winter or summer, his desire would slowly gravitate to a single optimum condition. Findings in the laboratory (59) indicate that this would probably be between 68 and 70 deg ET for most persons. According to the best information available from the Society's Laboratory and other studies, acclimatization requires from one to two weeks. A longer period is indicated for acclimatization to cold than is required for acclimatization to heat. The Laboratory has found from seven to ten days required for acclimatization to heat, either as a result of change in summer weather or for men working in a hot environment.

*Heat Dissipation to the Atmosphere.* In the application of summer cooling and air conditioning during the middle twenties, need developed for accurate data on the rate of heat dissipation from the human body to the atmosphere, and the differentiation of this heat into sensible and latent heat for different atmospheric conditions. These data were made available in a series of articles (20, 24, 67 and 68) presented to the Society and published in its Journal and in other journals. As viewed from the physiological point of view, this represents the greatest contribution of the Society's Research Laboratory and is accepted and used extensively by physiologists, particularly in studies dealing with the adaptation of man to different climatic and environmental conditions. These data are used universally by the air conditioning engineer in estimating the design cooling load resulting from occupancy.

*Air Conditioning in Industry.* The early development of comfort air conditioning found its greatest interest in attempting to provide ideal comfort in otherwise hot environments. First, the theater, and later, other audience halls, shops, restaurants, and other places of assemblage were cooled and conditioned. Usually the chief incentive was the box office receipts or appeal to prospective customers. The early and middle thirties witnessed a growing interest in the application of air conditioning to alleviate unbearable conditions in industry. The efficiency of the worker and labor laws were the major incentives.

Because of the unusual industrial surroundings, as contrasted with audience halls and other places of assemblage, and because ideal comfort is costly, it seemed necessary to base the standards to be maintained on the physiological responses of the worker and to permit conditions too warm for ideal comfort. A series of studies were initiated in 1937 which have continued until the present. The purpose in these studies was to maintain in the laboratory such environmental conditions as pertain to any given industry, to subject individuals

to those conditions whilst performing rates of work which pertain to the industry and to determine their physiological reactions and responses. The worker's body temperature, pulse rate, metabolism, respiration, blood pressure, vital capacities, degree of perspiration and feeling of warmth were studied. The results of a comprehensive study (49, 56, 62, 64, 69) establish the basis for other studies applying to different industries. As was to be expected, it was found that the effective temperature of the environment was the controlling factor, and that at least a fairly wide range in relative humidity is permissible as far as the worker's comfort is concerned.

*Present Conditions of Standards and Requirements.* Our knowledge concerning the physiological responses of persons to atmospheric conditions, as required for the application of air conditioning for human comfort, has kept pace with the professional and industrial requirements of the engineer. These findings have proven of great interest to the physiologist, who has only recently recognized it as a field in which he should be interested. The requirements and, therefore, the findings of the Society's Laboratory have not always involved the precision desirable for the establishment of academic scientific facts in which the physiologist is interested. Hence, there is a current interest in studies aimed to establish finer differentials between physiological response to small environmental changes. These studies will be of interest to the air conditioning engineer, and in some cases his practices may be altered in some small degree as a result thereof.

The present needs for further research as viewed from the point of view of the air conditioning engineer, include:

1. A better understanding of the relation between the comfort and health of a person, and the relative humidity of his environment, independent of the effective temperature. The results of studies at the Pierce Laboratory indicate (70) that the optimum relative humidity may be in the neighborhood of 70 per cent.

2. More accurate studies of environmental conditions, immediately above and below the sensible perspiration line, with a view of establishing the abruptness of this change and, therefore, indirectly the change in the slope of the effective temperature lines in this region of the Psychrometric Chart. Studies by the Society's laboratory would seem to indicate that any differences from present standards will be of academic rather than practical importance.

3. Further studies of the shock experienced upon entering or leaving a summer cooled and air conditioned space, from or to the hot outside weather. This is probably of greater importance to older people than those who served as subjects in the Society's studies.

4. Studies of physiological responses to heat, with a view of determining more accurately any harmful effects of exposure to moderately hot atmospheres, particularly the effect of exposure to heat during short periods of work or during the entire twenty-four hour day. Will sleeping in a comfortable environment compensate for an eight hour shift in hot industry?

*Military Interest in Man's Response to His Atmospheric Environment.* The violent tempo of the global war, requiring the highest degree of intelligence and performance on the part of man in handling mechanized equipment, has developed an interest in this subject far beyond anything heretofore experienced. Needless to say, the findings of the Society's Laboratory during the past twenty years have served a need in supplying a great deal of the basic data, as well as a vantage point from which the more specialized studies may proceed. However, the environmental conditions met with are much more extreme, and in many cases the need for precise data is more acute.

The demands made by the military establishments of our government upon the Society's Laboratory during the approach of the present war and the large

number of members of the Society who are expert in this subject and who have been called to serve in solving these problems testify to the community of interest between the military in fighting this war and the environmental-physiological research carried on by the Society during the past 20 years. For obvious reasons, the many lines of specialized interest in these studies and the courses which they are taking can not be discussed here. However, it may be said that many phases of the environmental-physiological subjects in which the Society and its research has been interested are being intensively investigated. Because of the urgent need for the results, these studies are seldom carried beyond the point of supplying the minimum information needed at the moment. With cessation of hostilities these findings will be available for civilian use, and it is certain that they will give a great impetus to the present day knowledge of the air conditioning engineer. Because of the spotty, as contrasted with comprehensive, nature of the studies, much will be required in correlating the results and filling in the gaps. As far as results of value to the air conditioning engineer are concerned, this will fall upon this Society and its Committee on Research. The many interests developed in this subject during the war by noted scientists, including the physiologist, aerologist, anthropologist, geographer, psychologist, and others, promise an intensified interest in the civilian application of air conditioning for human comfort after the war.

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**1246**

## A STUDY OF INTERMITTENT HEATING OF CHURCHES

By F. E. GIESECKE,\* COLLEGE STATION, TEX.

**T**HE FOLLOWING study is intended to aid in developing a systematic method of determining what percentage should be added to the capacity of a heating system designed for continuous heating in order that it might be used successfully for intermittent heating.

The study is difficult because the heat requirements of intermittently heated buildings vary with (1) the prevailing minimum outdoor temperature, (2) the interval between heating periods, (3) the construction, form, and size of the building, and (4) the length of the heating-up period.

For the first example to be studied, the small and simple church shown in Figs. 1, 2, and 3 was selected and the study limited to the auditorium. The study was based on the following assumptions:

1. The outdoor temperature was zero and remained constant at that temperature during the heating-up period.

2. The indoor air temperature was 32 F at the beginning of the heating-up period and was raised from 32 F to 70 F at a constantly varying rate during a 3-hour heating-up period.

3. During the heating-up period, heat was applied, at a gradually increasing rate, to the air within the building and transferred from the air to the walls, floor, ceiling, and furniture, except that portion which was necessary to raise the temperature of the air and to heat the air which filtered into the building by natural ventilation.

4. As the temperature of the air within the building was increased from 32 F to 70 F, the increase took place at a constantly varying rate, as shown by the dashed line of Fig. 4.

5. The exterior wall was of brick, 21 in. in thickness; its inner surface temperature was 35 F and its outer surface temperature, 1 F; so that, at the beginning of the heating-up period, the exterior wall was still losing heat to the air within the building and also to the air without the building.

6. The temperature gradient of the exterior wall was shown by the heavy, broken line of Fig. 4.

7. The windows were glazed with single glass; their area was 815 sq ft.

8. The floor was 7.2 in. concrete slab; its area was 3600 sq ft. One-fourth of the concrete slab was over a basement in which the air temperature was 40 F throughout the heating-up period; the remaining three-fourths of the concrete slab rested directly on the ground, the temperature of which was 60 F throughout the heating-up period.

9. The ceiling was of wood,  $\frac{3}{4}$  in. thick; its area was 43,000 sq ft.

10. The roof was of tin, on wood sheathing; its area was 43,000 sq ft.

11. The exterior wall was of brick; its inner surface area was 5300 sq ft.

12. The two outer doors were of wood, 2 in. thick; their area was 140 sq ft.

13. The interior columns and balconies were of wood; their weight was 18,000 lb.

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14. The furniture was of wood; its weight was 9000 lb.

15. The volume of air within the building was 102,000 cu ft; its weight was 7600 lb.

The study was based on the following reasoning:

1. As heat is applied to the air within the building, a portion of the heat is transmitted from the air to (a) the glass windows, (b) the wooden doors, (c) the wooden furniture, (d) the wooden columns and balconies, (e) the wooden ceiling, (f) the cement floor, and (g) the exterior brick wall.



FIG. 1. A SMALL AND SIMPLE CHURCH

2. The heat which is not so transferred is used to heat the air that filters into the building and to elevate the temperature of the air within the building.

3. The portion of the heat which is transferred to the furniture, columns, and balconies is consumed in elevating their temperatures.

4. The heat which is transferred to the windows, doors, floor, and ceiling is consumed partly in elevating their temperatures; the remainder passes to the outside of the building.

5. The portion of the heat which is transferred to the exterior masonry wall is absorbed by the wall and serves to elevate its temperature because a longer period of time than 3 hours is required for heat to flow through a 21-in. brick wall with the temperature differences assumed in this study.

6. It is sufficiently accurate, for the purpose of the study, to determine the flow of heat from air into a masonry wall, or other body, by means of the graphical method described in a recent paper.<sup>1</sup>

#### THE QUANTITY OF HEAT ABSORBED BY THE WALL

To study a variable flow of heat within a body, by the graphical method, it is necessary to divide the time into a series of equal intervals and to assume

<sup>1</sup> The Flow of Heat Through Walls, by F. E. Giesecke. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 441.)

that the flow of heat remains constant during each interval. For this study, a 3-hour heating-up period was selected, and the time was divided into 9 intervals of 20 min each.

The modified differential equation for the flow of heat in a body is

$$\Delta t \Theta = \frac{k}{cw} \frac{\Delta t}{(\Delta x)^2} \Delta^2 x \Theta \dots \dots \dots (1)$$

where

$\Theta$  = varying temperature in degrees Fahrenheit.

$\Delta t$  = increment in time, in hours.

$\Delta x$  = increment in distance, in feet.

$c$  = specific heat of the material.

$w$  = weight of the material, in pounds per cubic foot.

$k$  = thermal conductivity of the material, in Btu per hour per degree Fahrenheit, per foot.

$t$  = time, in hours.

$x$  = distance, perpendicular to the face of the wall, in feet.

To pass, by graphical methods, from one thermal gradient of the body to the next succeeding gradient, it is necessary to reduce Equation (1) to

$$\Delta t \Theta = \frac{1}{2} \Delta^2 x \Theta \dots \dots \dots (2)$$

This may be done by making

$$\Delta t = \frac{cw}{2k} (\Delta x)^2 \dots \dots \dots (3)$$

Having assumed  $\Delta t$  equal to 20 min or  $\frac{1}{3}$  hour, and assuming for the exterior brick wall,  $k = 0.5$ ;  $c = 0.2$ ; and  $w = 125$ , and substituting these values in Equation (3),  $\Delta x$  is found to be 0.115 ft or 1.4 in.

Since this study is concerned only with the flow of heat from the air into the masonry wall during 9 periods of 20 min each, it is sufficient to set off 9 distances of 1.4 in. each, beginning at the inner surface of the wall, as shown in Fig. 4. The purpose of this study is to determine the rate at which heat flows from the air into the wall during each of the 9 successive 20-min periods. The quantity of heat which flows from the wall into the outdoor air, during the same period, is irrelevant and, for that reason, it is not necessary to extend the successive thermal gradients farther than shown in Fig. 4.

The graphs of Fig. 4 were constructed as follows: The curved, dashed line represents the average air temperature, as the air is being gradually heated from 32 F to 70 F during the 3-hour period. The form of this curve was assumed so as to represent a gradual decrease in the rate of increase of air temperature; a slight change in the form of this curve would have no appreciable effect on the final outcome of the study.

The initial wall temperature gradient from 35 F on the interior surface to 1 F on the exterior surface was also assumed. The tangent to this gradient, at the inner wall surface, intersects the 32 F air temperature line at a point 3.64 in. from the wall surface. The slope of this tangent is  $(35-32)/3.64$  or 0.824 deg per inch; multiplying this slope, i.e., this thermal gradient at the inner wall surface by the thermal conductivity of the material, 0.5, for a gradient of 1 deg per foot, or 6, for a gradient of 1 deg per inch, determines the rate of heat flow from the wall to the air as  $6 \times 0.824$  or 4.94 Btu per hour per square foot. The rate of heat flow may also be found by multiplying the

temperature difference, wall surface to air, 35-32, by the film coefficient 1.65, namely,  $3 \times 1.65$  or 4.95 Btu per hour per square foot.

The location of the tangent to the temperature gradient at the wall surface and its use in calculating the rate of heat flow from the wall into the air, or from the air into the wall, may be explained as follows:

Referring to the sketch in Fig. 4, the quantity of heat flowing from the air into the wall is equal to the temperature difference, air to wall surface, multiplied by the film coefficient, or to  $a(t_a - t_s)$ ; it is also equal to the quantity of heat flowing, during the same period of time, from the surface of the wall into the layer of the wall

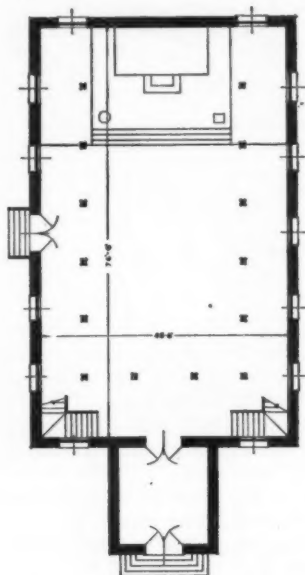


FIG. 2. FLOOR PLAN OF CHURCH

adjacent to the surface; this is equal to the thermal gradient at the surface multiplied by the thermal conductivity of the material or to  $k \, dy/dx$ ; hence,  $a(t_a - t_s) = k \, dy/dx$ . From the similar triangles,  $OF E$  and  $dx \, dy$ , it is evident that  $OF : FE :: dx : dy$ ; hence

$$OF = FE \, dx/dy = (t_s - t_a) \, dx/dy = k/a$$

In the present study, relating to a brick wall, the values of  $k$  and  $a$  are, respectively, 6 and 1.65, and  $OF$  is, therefore, 3.64 in. The point  $O$ , where the tangent to the thermal gradient intersects the air temperature line, is the *directing point*. Having drawn the tangent, the rate of heat flow into the wall is found by multiplying the slope of the tangent (in degrees per foot or per inch) by the corresponding thermal conductivity. For example, the slope of tangent 4 is 4.66 deg per inch; the rate of heat flow into the wall is therefore  $4.66 \times 6$  or 27.96 Btu per hour per square foot. This heat flow may also be

calculated by multiplying the corresponding temperature difference, air to wall, 60.2 — 43.4 by the film coefficient 1.65, which gives 27.92 Btu per hour per square foot. And difference in the values calculated by these two methods is the result of inaccuracies in the drawing or in the reading of the chart of Fig. 4.

The slopes of the 9 tangents shown in Fig. 4 are, respectively, 0.55-2.85-4.10-4.66-4.97-5.21-5.30-5.27 and 5.20 deg per inch.

The corresponding rates of heat flow, in Btu per hour per square foot, are 3.30-17.10-24.60-27.96-29.82-31.26-31.80-31.62 and 31.20.

The corresponding rates of heat flow, in Btu per hour for the entire wall, or for 5,300 sq ft, are 17,500-90,600-130,000-149,000-158,000-165,700-168,500-167,600 and 165,400. These values are shown in the chart of Fig. 7, in conjunction with the other rates of heat flow determined below.

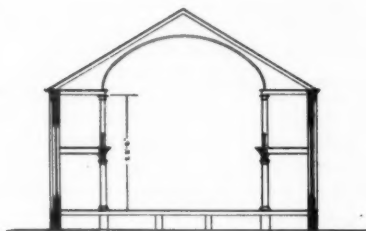


FIG. 3. SECTION OF CHURCH

#### THE RATE OF HEAT FLOW THROUGH WINDOWS AND DOORS

The thermal capacities of the windows and doors are so small that the increase in the quantities of heat stored in them during the heating-up period may be neglected in this study and the flow of heat calculated in the usual way.

The calculation may be simplified by combining the door areas with the window areas as follows: assuming  $U$ -values of 0.46 for the wooden doors and 1.13 for the single glazed windows, 46 sq ft of window surface will have the same heat loss as 113 sq ft of door surface; the 140 sq ft of door surface may therefore be replaced by 57 sq ft of window surface; this, when added to the 815 sq ft of window surface, gives an equivalent glass area of 872 sq ft.

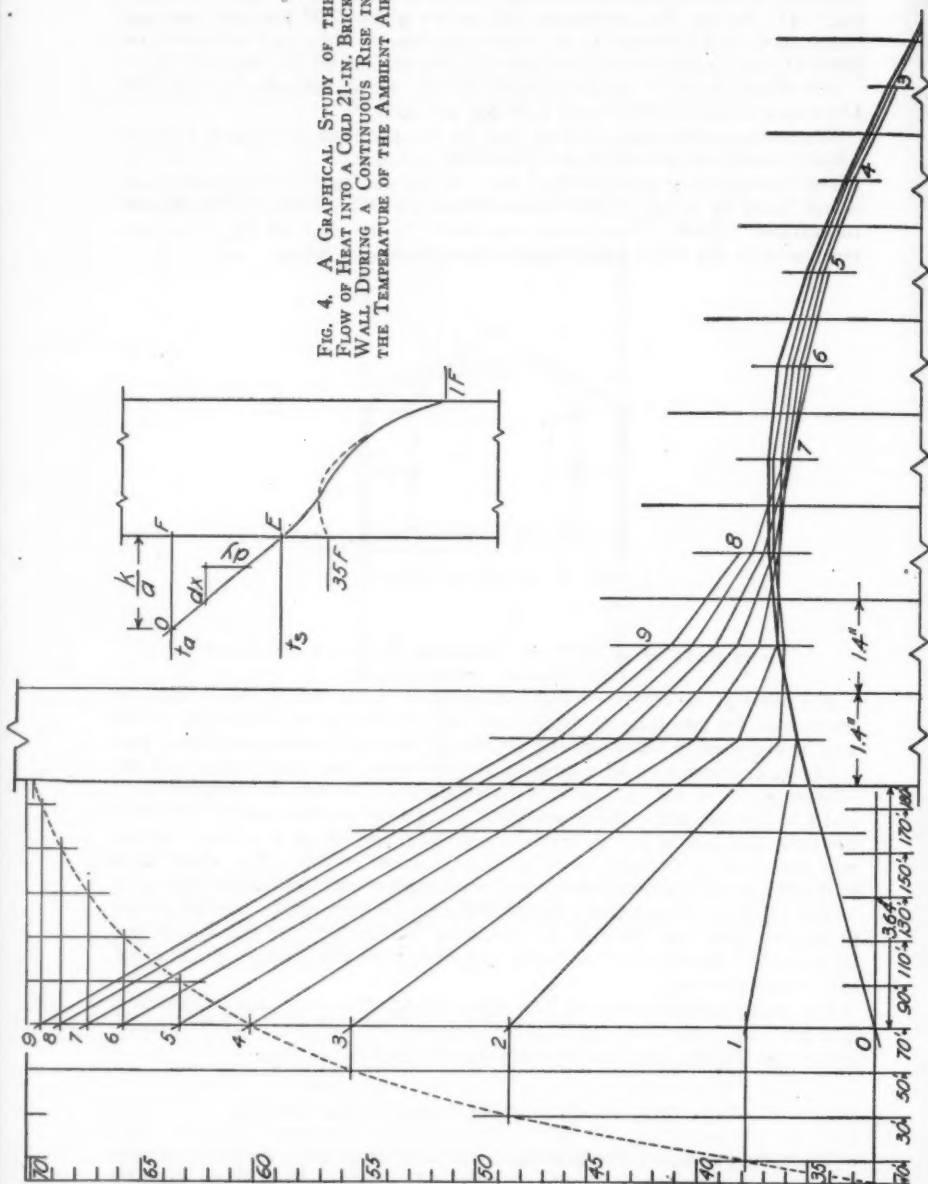
The mean air temperatures during the nine 20-min periods, into which the heating-up time was divided, as shown by the curved line on Fig. 4, are 37.9-48.6-55.7-60.2-63.4-65.9-67.5-68.7 and 69.6 while the outdoor air temperature remains at zero.

The corresponding rates of heat flow through the doors and windows in Btu per hour are, therefore, 33,000-42,300-48,500-52,500-55,200-57,400-58,800-59,800 and 60,600. These values are shown in the chart of Fig. 7.

#### THE RATE OF HEAT FLOW THROUGH THE CEILING

The ceiling and the roof sheathing are of thin wood; their thermal capacities are so small that it was considered sufficiently accurate to calculate the heat

FIG. 4. A GRAPHICAL STUDY OF THE FLOW OF HEAT INTO A COLD 21-IN. BRICK WALL DURING A CONTINUOUS RISE IN THE TEMPERATURE OF THE AMBIENT AIR



flow through them in the usual manner, using an over-all coefficient of 0.22. On this basis, the rates of heat-flow, in Btu per hour, during the nine successive 20-min periods are 37,000, 47,500, 54,400, 58,800, 61,800, 64,400, 66,000, 67,000 and 68,000, as shown in the chart of Fig. 7.

#### THE RATE OF HEAT FLOW INTO THE CONCRETE FLOOR SLAB

One-fourth of the floor slab was assumed to be over a basement and the remainder to rest directly on the ground. The two sections were studied separately. For the section over the basement it was assumed that the air in the basement was 40 F at the beginning of the heating-up period and that it remained constant during that period; also, that the flow of heat through the slab had become constant and that the temperature gradient in the slab, at the beginning of the heating-up period, was a straight line. Under these conditions, the temperature of the lower surface of the slab was 37.3 F at the beginning of the heating-up period and the temperature of the upper surface was 34.7 F. Consequently, heat was still flowing from the basement through the concrete slab into the auditorium.

The flow of heat through the slab, during the nine 20 min periods, was studied by means of a chart like that shown in Fig. 4, and it was found that the nine successive, upper surface temperatures were 35.2, 36.0, 39.0, 40.9, 42.5, 44.0, 45.3, 46.5, and 47.6 F. By subtracting these temperatures from the corresponding air temperatures and multiplying the differences by 1.56—the film coefficient—it was found that heat was flowing into the slab at the rates of 4.2, 19.7, 26.1, 30.2, 32.6, 34.2, 34.7, 34.7, and 34.4 Btu per hour.

By multiplying these rates by the area of the slab, 900 sq ft, it was found that heat was flowing from the auditorium into the concrete slab above the basement at the rates of 3,800, 17,700, 23,500, 27,200, 29,300, 30,800, 31,200, 31,200, and 31,000 Btu per hour; these values are shown in the chart of Fig. 7.

For the section of the floor slab resting directly on the ground, it was assumed that the temperature of the ground was 60 F and remained constant during the heating-up period; also, that the flow of heat had become constant and that the temperature gradient in the slab was a straight line at the beginning of the heating-up period. The flow of heat through the slab was studied as described previously and it was found that the nine successive, upper surface temperatures were 37.9, 41.7, 44.8, 47.4, 49.6, 51.6, 53.4, 54.9, and 56.3 F, and the rates of heat flow into the slab were 0.0, 14.1, 17.0, 20.0, 21.6, 22.3, 21.9, 21.6, and 20.8 Btu per hour per square foot, and that heat was flowing from the auditorium into the concrete slab resting directly on the ground at the rates of 0, 38,100, 46,000, 54,000, 58,400, 60,200, 59,100, 58,400 and 56,200 Btu per hour. These values are shown in the chart of Fig. 7.

#### THE RATE OF HEAT FLOW INTO THE WOODEN FURNITURE, COLUMNS, AND BALCONIES

Assuming the weight of wood as 45 lb per cubic foot, the thermal conductivity at 0.12 Btu per hour per degree Fahrenheit per foot, and the specific heat as 0.5, the distance interval  $\Delta x$  must be 0.695 in. if the time interval  $\Delta t$  is to be 20 min. Evidently, a distance interval of 0.695 in. is too large for use in studying heat flow in thin wooden members like those used in the construction of furniture, floors, etc. For those cases the distance interval should

not exceed  $\frac{1}{8}$  in. and, consequently, the time interval should not exceed 6.4 min; using these values to study heat flow during a 3-hour period would necessitate drawing about 55 thermal gradients instead of the nine shown in Fig. 4; this would be very difficult and it appears practically impossible to study the flow of heat in thin wooden members by the usual graphical method.

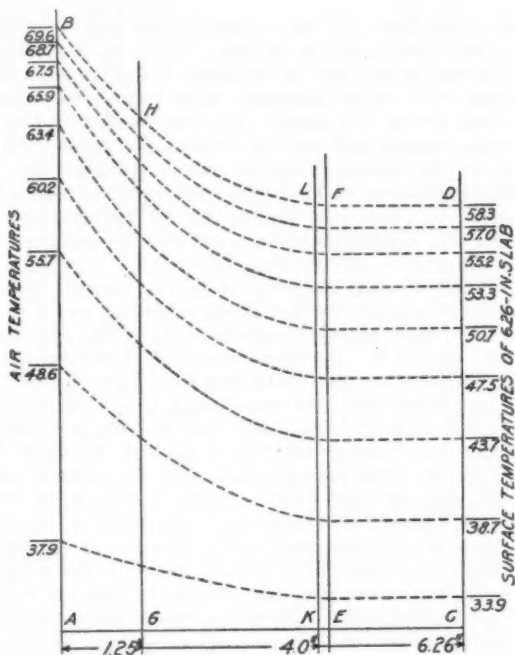


FIG. 5. AN APPROXIMATE METHOD OF DETERMINING THE SURFACE TEMPERATURES OF THIN SLABS OR OF THIN WALLS

For that reason an approximate method was adopted which is based on the following reasoning.

If a large wooden slab were cooled to 32 F and placed in the auditorium at the beginning of the heating-up period, it would absorb heat from the surrounding air and its surface temperatures would increase as the temperature of the surrounding air increases, but its increase in surface temperature would lag behind that of the air temperature.

If the thickness of the slab were reduced, the lag in the increase of surface temperature would also decrease and as the thickness of the slab approaches zero the lag would also approach zero. A sheet of wood so thin that its thermal capacity is negligible would always be at the temperature of the surrounding air.

This is illustrated in Fig. 5. The temperatures shown there on the Line *AB* are the mean air temperatures of the nine successive 20-min periods derived from Fig. 4. The temperatures on the Line *CD* are the corresponding surface temperatures of a slab of wood 6.26 in. thick, determined as shown in Fig. 6. The temperatures on Line *EF* are the corresponding surface temperatures of

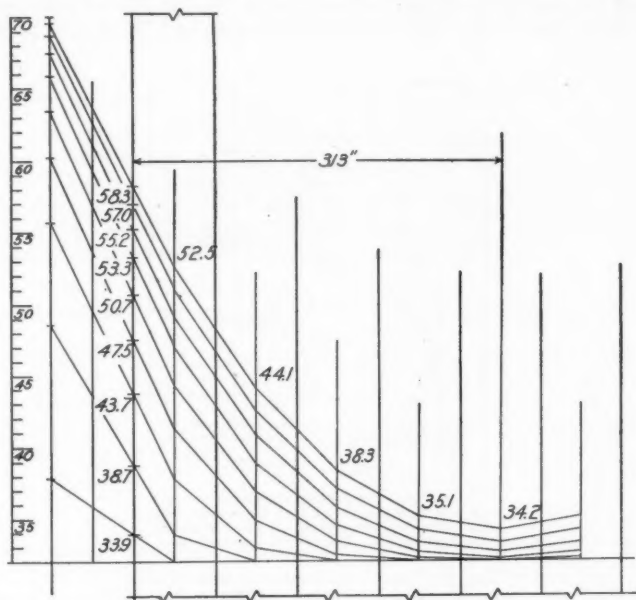


FIG. 6. A GRAPHICAL STUDY OF THE VARYING SURFACE TEMPERATURE OF A 6.26-IN. WOODEN SLAB DURING A CONTINUOUS RISE OF THE TEMPERATURE OF THE AMBIENT AIR

a slab of wood 4.17 in. thick, determined in a manner similar to that shown in Fig. 6.

The differences between the temperatures shown on Line *CD* and those shown on Line *EF* are smaller than was expected; both were determined as accurately as possible.

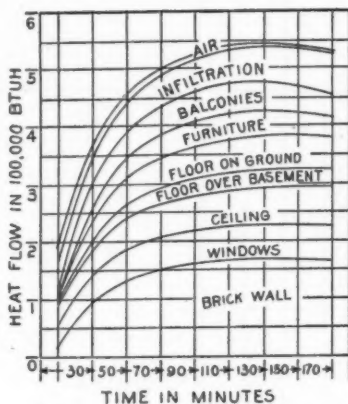
The dashed lines are intended to show the probable variation in the nine mean surface temperatures as the thickness of the wooden slab is reduced from 6.26 in. to zero.

If it is assumed that the principal parts of the furniture are built of  $1\frac{1}{4}$  in. boards, the successive surface temperatures might be determined by drawing a line *GH* parallel to and at a distance of  $1\frac{1}{4}$  in. from line *AB* in Fig. 5.

In this manner the successive temperature differences, air to surface of wood, were found to be 1.7, 4.8, 6.0, 6.7, 6.9, 6.7, 6.5, 6.3, and 5.9 deg. The weight of the furniture was estimated to be 9000 lb.

The furniture was, therefore, replaced by 1920 sq ft of  $1\frac{1}{4}$  in. boards; and the heat absorbed by these boards found by multiplying 1920 by  $2 \times 1.65$ , since heat flows into the boards from two sides, and by the successive temperature differences listed, to be 10,800, 31,000, 38,000, 42,500, 43,800, 42,500, 41,200, 40,000 and 37,400 Btu per hour. These values are shown in Fig. 7.

The heat flow into the columns and balconies was found by assuming that it would be equal to the heat flow into a 4 in. wooden slab of equal weight, 18,000 lb, i.e., into a 4 in. slab having a surface area of 1200 sq ft on each side.



Vertical distances between the lines show rate of heat flow into the part of the construction indicated between the lines. The top line is the sum of the increments for each part.

FIG. 7. THE SUMMATION OF THE NINE RATES OF HEAT FLOW, OR HEAT LOSS, DURING THE THREE-HOUR HEATING-UP PERIOD, DIVIDED INTO NINE 20-MINUTE INTERVALS

The line *KL* in Fig. 5 was drawn parallel to and 4 in. from line *AB* and the temperature differences, air to wood surface, found to be 4.0, 9.7, 12.0, 12.6, 12.7, 12.7, 12.2, 11.7, and 11.1 deg. By multiplying these temperature differences by  $2 \times 1.65 \times 1200$  the following rates of heat flow were found: 15,800, 38,400, 47,400, 49,900, 50,300, 50,300, 48,400, 46,400, and 44,000 Btu per hour. These values are shown in Fig. 7.

The following interesting information may be obtained from the chart of Fig. 6:

1. The air temperature rose from 32 F to 69.6 F during the heating-up period.
2. The surface temperature of the wooden slab rose from 32 F to 58.3 F.
3. The temperature of the wood in the center of the slab rose from 32 F to 34.2 F.
4. The average rise of the temperature of the wooden slab was about 9.6 deg.

5. The quantity of heat absorbed by the wooden slab was about  $9.6 \times 3.75 \times 3.13 \times 0.5$  or about 56 Btu per square foot of exposed surface, during the heating-up period, or about 18.7 Btu per hour.

6. The average temperature difference, air to slab surface, was, therefore, about  $18.7/1.65$  or about 11.3 deg.

7. The temperature differences, air to slab surface, shown in the chart of Fig. 6, range from about 4 deg to about 11.3 deg; their average value is about 9.9 deg, which checks fairly well with the calculated value based on the quantity of heat absorbed, as determined by the graphical method.

#### RATE OF LOSS OF HEAT RESULTING FROM INFILTRATION

It was assumed that air filtered into the auditorium at the rate of 50,000 cfm, about one-half the volume of the auditorium; also that the average weight of the air was 75 lb per 1000 cu ft and that its specific heat was 0.25. On this basis, the resulting rates of heat loss for the nine consecutive 20-min periods were 35,500, 45,500, 52,100, 56,400, 59,300, 61,600, 63,100, 64,400, and 65,100 Btu per hour. These values are shown in the chart of Fig. 7.

#### RATE OF HEAT ABSORPTION BY THE AIR WITHIN THE AUDITORIUM

Assuming the volume of air within the auditorium as 102,000 cu ft, its weight as 7,600 lb, and its specific heat as 0.25, the nine successive rates of heat absorption, based on the rates of temperature rises listed above, are 61,500, 49,000, 41,600, 21,100, 14,800, 11,400, 7,400, 5,700, and 4,000 Btu per hour. These values are shown in the chart of Fig. 7.

#### MAXIMUM RATE OF HEAT FLOW

According to this study, the maximum rate of heat flow was about 543,000 Btu per hour. According to the usual method of calculating heat losses, the maximum rate of heat flow would be about 314,000 Btu per hour, as shown by the following calculation. Consequently, the capacity of the heating plant should be about 543,000/314,000 or about 1.73 times as large for the intermittently heated church as for the same building heated continuously, under the conditions assumed for this study (see Table 1).

TABLE 1—HEAT LOSS CALCULATION

SURFACE	AREA	COEFFICIENT	TEMPERATURE DIFFERENCE	HEAT LOSS BTU PER HOUR
Walls.....	5300	0.20	70	74,200
Glass.....	815	1.13	70	64,500
Doors.....	140	0.46	70	4,500
Ceiling.....	4440	0.22	70	68,500
Floor.....	900	0.53	30	14,300
Floor.....	2700	0.83	10	22,400
Infiltration, 50,000 cu ft.....			70	65,600
				314,000

## CONCLUSIONS

A building heated intermittently requires a heating system of considerably larger capacity than the same building heated continuously.

The subject of intermittent heating is of sufficient importance to justify additional studies to determine the effects of outdoor temperature, interval between heating periods, length of heating-up period, and the form, size, and type of building.

## ACKNOWLEDGMENT

The author is indebted to E. K. Campbell, President, E. K. Campbell Heating Co., Kansas City, Mo., for much valuable aid in the preparation of this paper and particularly for the choice of the determination of the highest rate of heat in-put during the heating-up period, rather than the total quantity of heat in-put during that period, as the prime object of the study, and to Dr. H. Grober, Director, Charlottenburg Experiment Station of Heating and Ventilating, for the selection of an assumed variable indoor air temperature as the basis for the study of intermittent heating.

## DISCUSSION

E. K. CAMPBELL, Kansas City, Mo. (WRITTEN): Dr. Giesecke's continued interest in the matter of the intermittent heating load for churches and other intermittently heated buildings has been of much comfort to me in view of the fact that through a number of years, I have been unable to interest the Society in that particular subject. I hope the two presentations he has made will increase the interest and bring a realization that the Society is overlooking the lack of knowledge in this particular field, and the consequent large number of poorly heated churches and other buildings which are not heated continuously.

Our Society is very proud of its original research; the original studies of problems where information was lacking, and the results it has been able to show. Here is another opportunity.

Many years ago, the designing of church heating systems brought home to me that this was a vital problem. We had neither the equipment, the time, nor the mathematical facilities to go into this problem scientifically. We did reach the conclusion after much study that the heat absorbed by the cold material was the determining factor, and that calculations must be based on the maximum combination of the heat flow into the cold material and the heat flow out of the building in the normal manner. We recognized that there would be a changing ratio, and that at some point in the heating up period, there would be a maximum combined figure that would determine the minimum capacity of the heating plant required to satisfactorily heat a cold building in a given length of time. As the author has pointed out, the load depends not on the entire amount of heat absorbed by the cold material in the heating up process, but on the maximum amount absorbed and transmitted at any one time within that period.

If, at that time, the heating plant has the capacity to supply more than all of the heat which is absorbed and transmitted, then the heating up curve will continue to rise and will reach the desired temperature in a fairly short time. Dr. Giesecke, in his Fig. 4, has assumed a curve showing that the heating plant has this additional capacity, so that the heating up curve does not flatten off. In Fig. A, I am repeating his curve as Curve No. 1; Curve No. 2 is a similar curve up to the point of maximum combined absorption and transmission, and is based on the assumption that the heating

plant can supply the exact amount of this combined absorption and transmission. Hence, when the curve reaches the point where the combined amount is equal to the total capacity of the plant, the curve tends to flatten off and will rise only after some of the absorption load is satisfied, and after a longer period will bring the church up to the desired temperature. Curve No. 3 is based on the assumption that the heating capacity of the plant is exactly the calculated maximum heat loss of the building. Hence, as shown, the curve levels off at the lower level, the level of the maximum capacity of the plant, and will continue to run off nearly level indefinitely if the

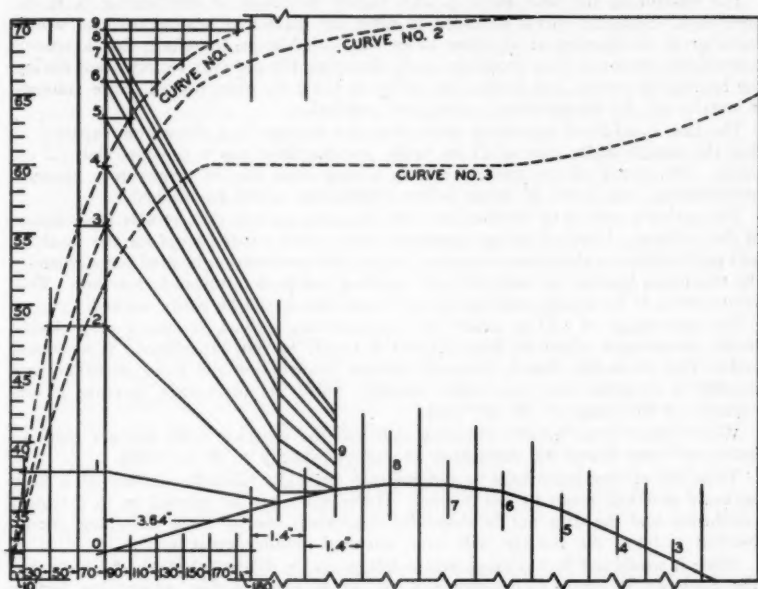


FIG. A. CURVES SHOWING HEATING EFFECT OF: *a*. SUFFICIENT CAPACITY TO HEAT IN 3 HOURS; *b*. CAPACITY EQUAL IN AMOUNT TO MAXIMUM DEMAND; *c*. CAPACITY EQUAL IN AMOUNT TO ESTIMATED NORMAL HEAT LOSS

weather is sufficiently cold and the building has been cold long enough so that there is the maximum absorption of heat by the material.

This illustrates the need for a plant of much larger capacity than is indicated by the normal heat loss figures on any intermittently heated building.

It is worthwhile to note that even though the plant has the capacity to bring the temperature up to 70 deg in three hours, according to Dr. Giesecke's assumed curve, the curve is still rising, the building has not reached an equilibrium, and will not for a number of hours. It follows, therefore, that if the heating plant has less capacity, either of the order of Curve No. 2 or of Curve No. 3, the time required to reach not only 70 deg, but equilibrium, will be proportionately greater.

An analysis of the figures and curves shows that the maximum amount of heat was required at about the seventh of the nine periods. From then on, the absorption decreased and the transmission to the outside increased, but the total decreased. Equilibrium was being approached.

Another feature of importance is the radiant cooling effect, if the surrounding material is not warmed and consequently receives radiant heat from the person without radiating a sufficient amount back. It would, of course, not be possible to heat the air without putting some heat into the material, but if the air temperature is brought up rapidly, the material will be heated much more slowly so that this radiant cooling effect will prevail. Hence, comfort cannot be established until the surface temperature of the cold material is raised to where its absorption is not so rapid nor so great.

The control of the heat input is also highly important if overheating is to be prevented. Sufficient direct radiation to meet the maximum heating up load would cause great overheating at all other times. It would seem, therefore, that a remote controllable source of heat should be used, obtaining 100 per cent recirculation during the heating up period, and having the ability to bring in, when needed, large volumes of outside air for temperature control and ventilation.

The case considered has rather more than the average heat absorption capacity in that the outside walls were of 21 in. brick, and the floor was a concrete slab 7.2 in. thick. The center of the material shows a very slow rise in temperature, thereby necessitating long hours of firing before equilibrium would be reached.

The author's method of dividing the time into nine periods greatly aids the solution of the problem. I believe he has presented and worked out the basis for the involved and multitudinous calculations necessary to put this information in final form whereby the maximum heating up load of a cold building can be determined in advance. This information, to be useful, must be in such form that it can be easily applied.

The percentage of 1.73 at which Dr. Giesecke has arrived, compares very closely to the percentages which we have arrived at purely by our experience. If he would add to this particular church, fireproof interior construction and many partitions and possibly a concrete roof, we would normally raise the percentage increase in the capacity of the plant to 100 per cent.

Where there is no interior concrete slab and the exterior walls are not quite so heavy, we may lower the percentage of excess capacity to 50 per cent.

Thus, all of our experience would indicate that Dr. Giesecke arrives at a very accurate practical result by his method. If the work can be carried on to a logical conclusion and the data put in shape for convenient use by those designing church heating systems, the Society will have rendered another great service.

Since it would not be too expensive to follow up Dr. Giesecke's method I hope that the Society's Research Committee will see fit to take up this subject for further development at an early date.

E. G. SMITH, College Station, Tex. (WRITTEN): This paper has made an important contribution to the art of intermittent heating because the author has called attention to the fact that during unusually cold periods, the interior portions of an exterior wall are practically certain to be at a temperature somewhere near the temperature of the air in the interior of the building itself. This has a very direct bearing on the computation of intermittent heating loads by the absorbance method described by the writer in a previous paper.<sup>2</sup> As described in this paper and also more fully in Bulletin No. 62 of the Texas Engineering Experiment Station, the heat requirements for an intermittently heated building come out too high. It was already known that the surface coefficients used were too high for the transient conditions that exist during the period of rising air temperature. Dr. Giesecke and Mr. Campbell have made it clear that the *steady state* load will certainly be negligible and may even be negative. As a result, the *steady state* part of the load can be omitted entirely. This makes the computation of an intermittent load for a given wall just as simple as the computation of the *steady state* load for the same wall. The only difference is that a table of absorbances, instead of a table of conductances, must be used.

<sup>2</sup> A Method of Compiling Tables for Intermittent Heating, by Elmer G. Smith (A.S.H.V.E. Journal Section, *Heating, Piping & Air Conditioning*, June 1942, p. 386).

W. H. CARRIER, Syracuse, N. Y.: I believe it has been the usual practice to allow about 50 per cent excess heating capacity for intermittently heated or cooled buildings. When the desired optimum temperature is approximated, two-thirds of the capacity of the equipment will still be required to maintain equilibrium. This, alone, does not determine the time required to reach equilibrium which will differ greatly with different building constructions.

You can approach the problem of intermittent heating and cooling by either of two methods. First, to assume the existing capacity which first cost and engineering considerations may dictate and then calculate the time required, or second you may proceed as Dr. Giesecke has done, to assume the time required to reach the desired state and then calculate therefrom the capacity of equipment required.

With a brick wall or other material having approximately the same diffusivity as brick, that is, the specific heat divided by the product of density and specific heat is the same, it will require a heat wave about thirteen hours to travel through a 13-in. wall while it will only take one-quarter that time, or three and one-quarter hours for the same wave to travel through a 6½-in. wall. For this reason, there is not much difference in the size of apparatus required to heat up a building with either wall in a period under six hours. The advantage will lie with the heavier wall only, I believe, in the longer heating up period.

A very interesting problem of this nature arises in the cooling of mines. Due to the fact that the mine tunnel or drift is surrounded by a relatively infinite mass of warm rock, equilibrium is theoretically never reached but over 70 per cent of the ultimate cooling is done within the first year of continuous cooling.

W. A. DANIELSON, Memphis, Tenn.: I had some experience once in trying to heat for one winter a church with solid masonry walls. That caused me to look at these curves of Dr. Giesecke's, to see whether or not they actually worked out with our experience. I have a question that I would like to have him answer for me.

When you heat up your furnace and keep it fired correctly, it has a fairly steady Btu output. Where do the Btu's go? In 30 min you put out that many Btu's—the output (see Fig. 7) is about 370,000 Btu. In twice that time the output shown is 490,000 Btu's. The difference 490,000 minus 370,000 that is 120,000 Btu's, but the furnace is putting out heat at twice that rate, which should be 740. Where does that extra heat go?

L. T. AVERY, Cleveland, O.: My contact with heating of churches was in a specific church, and I think now it ought to be obvious that in a church the heating engineer is given one of the worst possible heating problems. If you think in terms of the heat absorption of these thick masonry walls the cost of heating then is that due to heating the walls and this has no significance so far as the church service is concerned. We had experience with a simple church with frame walls which very quickly heated and from the standpoint of speed of heating up was ideal. But the cooling effect of the cold walls was very disagreeable to the people near the walls. It was a small church heated by warm air, and it was not considered practicable to try to heat the walls by means of any radiation.

As a heating engineer, I was torn between the problem of adding insulation to try to make a warmer wall and make a slower heating-up period, or of trying to put on the wall some reflective material which would make a warmer wall without increasing its mass. We compromised by using conventional wall fill insulation. It was a poor solution. I think in any church of this kind the question of weight of insulation, and type, are probably of equal significance with the quantity of heat required.

H. M. HART, Chicago, Ill.: I think we have an opportunity for Mr. Close to do a little more work on his law of diminishing returns. Some time ago Mr. Close presented a paper<sup>2</sup> on the law of diminishing returns in connection with insulation. I

<sup>2</sup> Diminishing Effectiveness of Successive Thicknesses of Insulating Materials, by Paul D. Close. (A.S.H.V.E. JOURNAL SECTION, Heating, Piping & Air Conditioning, March 1940, p. 191.)

think it might well be applied to the heating of churches. I am particularly interested in the wall temperature, at what point the outside wall temperature reached a comfortable degree where the heat loss from the people sitting adjacent to the walls was not too great. That is the factor of comfort in churches.

It is true that a church is about the most complicated heating problem that we encounter, because of the intermittent heating and because of the shape of the structure. We have very high ceilings, and many times a lot of glass surface that is high up in the air and causes down-drafts. In the Research Residence, Urbana, Ill., heated by a direct hot water system, it takes about four hours at zero outside temperature to raise the temperature 7 deg in the house, and it is a well-insulated house.

Now, there is not much insulation in a church, but we do have masses of very cold walls. In my own experience we have felt that it is not unreasonable to ask the church custodian to start bringing the building up to temperature at least 10 hours before the time of occupancy. In some structures we do not obtain the wall temperatures we would like to have in that ten-hour period even though it is common practice to add about 50 per cent to the heating capacity for such a structure due to intermittent heating.

I have not studied this paper, so that I do not know whether the inside wall surface temperatures were given—whether or not they were reached after that period of two hours; but it takes two hours to raise a residential structure a few degrees, and when you are trying to raise a church structure as much as 40 deg I am inclined to think that it will take a very excessive heating installation, and one that is hard to control after the desired temperature is reached, especially during the average winter weather, which requires only about half the load encountered during the most severe weather.

R. G. VANDERWEIL,<sup>4</sup> Waterbury, Conn.: It is worthwhile to mention that to provide for comfort conditions in this church, we must not necessarily heat all the walls.

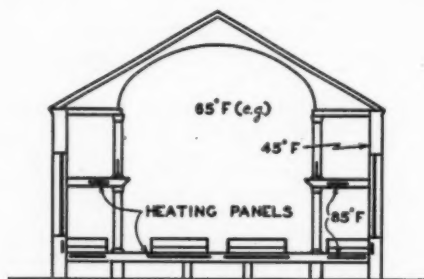


FIG. B. SMALL PANEL SECTIONS NEAR THE PEWS RAISE THE LOCAL MRT AND PROVIDE FOR COMFORT

In terms of the wall-temperature curves, I would imagine that comfort may be found at some point of the upper half of the rising branch of this curve, if we insert supplementary radiation panels.

So far, I have not been concerned with the problem of heating churches, but within a study of radiant heating I have given some thought to similar problems and think a solution as shown in Fig. B seems rather feasible because of the saving of fuel and shortening of the initial heating period:

<sup>4</sup> Project Engineer, Chase Brass & Copper Co.

If warm air is delivered into a *room of excessive height* which is not heated to full temperature, it will rapidly cool down, but even then comfort may be found if small wall and floor sections, located in close neighborhood to the occupants, are heated to a temperature well above the comfort temperature.

To illustrate the combined action of extremely different air and radiant temperatures, I would like to cite Dr. Carrier's example of the mine. In this mine with its 140 F wall temperature, we could—at least theoretically—still find comfort if we supply air of freezing temperature—just as the skier finds comfort up in the mountains, when taking off his shirt and exposing himself to the sun radiation.

According to the same principles, comfort may be found in the church if small wall sections of high temperature—which may be heated up within a short period, are provided in close neighborhood to the occupants, even if the temperature of the bulk of the walls and the air temperature are low.

DR. GIESECKE: I am pleased to see such interest in this paper—it is greater than I had expected or even hoped for.

Mr. Campbell's contribution is very valuable, because he has had so much experience in church heating. Dr. Carrier's comments are valuable and accurate as they always are.

In heating a church, one must either select the time required for the heating-up period and determine the excess capacity of the plant, or assume the excess capacity of the plant and then determine the time required for the heating-up period.

In that connection, however, the wall surface temperature must be taken into account.

General Danielson's question is one of which I thought long before he asked it. I did not know the answer, and I do not know it now. If the furnace emits heat at a uniform rate, then, of course, this curve which represents the total heat in-put must be a straight horizontal line. If the total heat in-put is correctly represented by the curved line in the figure, it follows that the furnace does not emit heat at a uniform rate or that the air in the building or a portion of the air is heated more rapidly than assumed in the study.

GENERAL DANIELSON: I think the answer is that air stratification is obtained as it is in hangars, and that as the air temperature at the top is quite high, the heat absorbed by the air is greater than supposed and consequently the heat losses in the higher parts are higher.

DR. GIESECKE: That assumption complicates the problem even more. It is quite complicated now.

MR. CAMPBELL: In heating buildings with high ceilings you must handle larger volumes of air, at lower temperatures, in order to avoid the high temperature at the ceiling. In the Butler University installation we handled the air at such a rate and the cubical contents were so great in proportion to the heat loss that a 25-deg temperature rise of the air passing through the furnace did the job with a 12-deg air temperature variation in a 90 ft height. A large air movement in proportion to the heat capacity of the furnace is required. The furnace does not give out heat at a constant rate as the rate of combustion is reduced as the heat requirements are satisfied.

DR. GIESECKE: It will be noticed that when the air temperature has reached 70 deg at the end of the three-hour period, the wall surface temperature is still only about 51 deg, and the mean radiant temperature near the floor must be below 50 deg, because the window temperature is lower than 50. A mean radiant temperature of 50 F with an air temperature of 70 F does not give comfort conditions unless the persons within the church are clothed more heavily than was customary in the tests by means of which it was found that comfort conditions were attained when the average of the mean radiant temperature and the air temperature was about 70 F.

Referring to Mr. Vanderweil; his statements are correct, but he is discussing radiant or panel heating instead of warm air heating which is under consideration at this time. If the lower portions of the walls of a church are provided with heating panels comfort conditions can be attained more readily than without such panels.

I would like to make one additional statement. I was in England during 1929 and '30, and was told by one of the engineers that in one of the English churches the heating was so unsatisfactory that the use of the church was practically abandoned until it was suggested the church be heated continuously instead of intermittently. The suggestion was adopted and the results were found quite satisfactory.

This indicates that there cannot be a very great difference between the cost of heating a building continuously and that of heating it intermittently and that there can be only a relatively small saving in the cost of heating by night set-back.



**1247**

## THE RESISTANCE TO HEAT FLOW THROUGH FINNED TUBING

By W. H. CARRIER\* AND S. W. ANDERSON,\*\* SYRACUSE, N. Y.

**T**HE PAST twenty years have witnessed a great advance in the art of heating and cooling air. Prior to this period, prime surface was employed almost exclusively. Today, prime surface has been largely replaced by extended or finned surfaces. This advance in the art has been accelerated by the availability of non-ferrous metals of relatively low thermal resistance but at the same time highly resistant to external and internal corrosion. While technical advances have made this change practicable, the engineering and economic stimulus for the substitution of non-ferrous metals with extended surfaces has been largely provided by the growth of air conditioning with its more critical requirements for heat transfer surfaces—types that would be compact, efficient, and at the same time highly resistant to corrosion in handling moisture-laden air.

While surfaces of this character had long been successfully used in automobile radiators, it was not until a little over twenty years ago that this type of construction was successfully adapted to the art of heating, ventilating and air conditioning. It at once filled a great need in this art which was then at the very beginning of its period of rapid growth. Thus we see that the timing of this development was most auspicious both for the commercial development of such surfaces and for its indispensable assistance in the development and in the commercial exploitation of air conditioning.

From the foregoing it will be seen that the technique of design of finned heat transfer surfaces, particularly as employed in heating and ventilation, is of very considerable commercial importance as well as technical or engineering interest. These design factors are also of great practical importance in the design of fins for air-cooled airplane motors as well as in many other industrial applications.

In any heat transfer surface, there are three factors which determine the rate of heat exchange:

1. The resistance to heat flow through the film of the heating or cooling fluid to the conducting metal surface.
2. The resistance to heat flow of the metal itself from the inner surface to the outer surface.
3. The resistance to heat flow from the outer metal surface to the air stream which is to be heated or cooled.

In the use of prime surface in heating or cooling of air, the first resistance between fluid and wall is relatively small and the resistance of the metal

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wall itself is wholly negligible. With finned surface, however, the finned surface on the air side may be from 10 to 20 times that of the prime surface in contact with the heating or cooling fluid. Furthermore, the length of the path of heat flow through the metal itself is many times that through the prime surface. From this it may be seen that for a given rate of heat flow, resistances (1) and (2), which in previous practice were insignificant, now have an importance almost equalling that of the resistance on the air side. In fact, in many applications the temperature drop from surface to air is not more than one-half of the total temperature drop, while a temperature drop

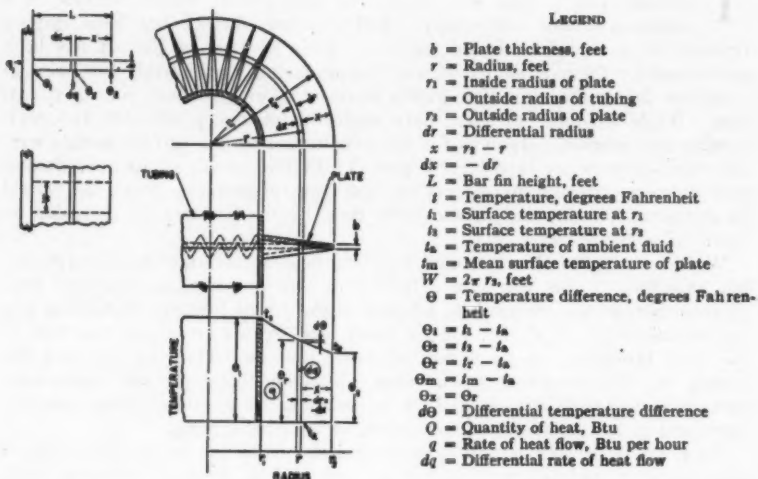


FIG. 1. BAR FIN AND EQUIVALENT FINNED TUBE

of more than about 70 per cent on the air side can hardly be considered the best economic practice.

For many years, manufacturers of finned heating and cooling surfaces have had data that was applicable in determining the performance of a particular design of finned surface for both the heating and cooling of air, including condensation of moisture. However, little data is available in usable form for calculating the effect of dimensional changes in the surface itself. It is the purpose of this paper to provide design data in a readily usable form which will facilitate optimum design for this type of heater for any service.

The principal design factors involved are *first*, the ratio of prime surface to extended surface, which in turn involves the spacing of the fin and the lineal extension of the fin beyond the prime surface. *Second*, the thickness of the fin and the conductivity of the metal employed.

Except in certain special applications where weight and space are at a premium, the *only* proper basis of comparison of two heat exchange designs

is the relative cost of two designs that give identical performance. There are two criteria of performance which must be the same for both.

1. The two units must handle the *same quantity* of air at the *same pressure drop*. (The face area and face velocity may be quite different.)
2. The ratio of final to initial temperature difference must be identical.

The factor which is most difficult either to calculate or to determine experimentally is the average conductance of the fin itself. In many designs, particularly that of the flat disc or plate fin type of extended surface, the metal resistance is the determining factor in optimum design.

The principal contribution made by this paper is a temperature ratio chart, Fig. 5, which determines the resistance of any design of flat plate surface including (1) any diameter of tubing, (2) any spacing of tubing, and (3) any thickness or material of the fins.

The resistance to air flow and the corresponding air side heat transfer factor can be calculated independently to only on approximate accuracy. Final values for the performance in these two respects must be checked by tests. It is not the purpose of this paper to discuss this phase of the design as affecting the performance, but a correct method of comparison of two different types of heaters is developed and a formula for such comparison is given.

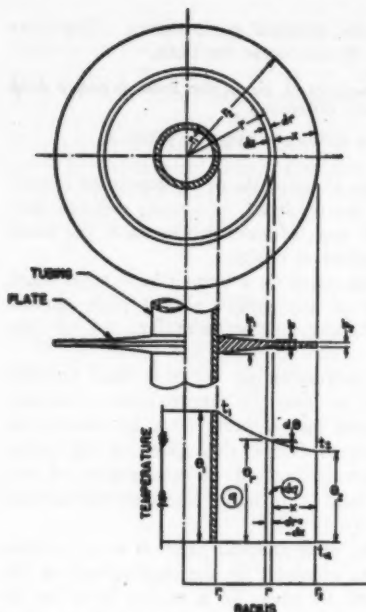
The general method of approach to the mathematical solution of a problem of metal resistance and the final results obtained by the application of the mathematical equations thus deduced will be given in a usable form in the body of the paper. The several mathematical solutions involved will be given in appropriate appendices for the convenience of those few who care to give the subject more critical study.

At least one of these solutions has previously been made and published by others.<sup>1</sup> Such derivations, however, have applied primarily to a bar fin of uniform thickness. This has a relatively simple mathematical solution while the mathematics of the plate fin transfer, to the authors' knowledge, has not been adequately covered as attempted in this paper. The material and calculations contained in this paper were made under the authors' direction some years ago and are now being taken out of the *mothballs*.

Commercial finned surfaces may be divided into three classifications, each of which has its own mathematical solution. In each of these forms, certain simplifying assumptions must be made in order to permit an accurate mathematical solution. In general, these assumptions are not widely at variance with practical results and are at least of comparative accuracy, which is the essential thing. The principal assumptions are:

1. That there is a known and fixed surface temperature at the tube.
2. That the coefficient of heat conduction from air to surface is constant for all points on the surface. This is only approximately true as the turbulence varies the conductance. Also, in condensing moisture from the air, the area nearest the tubing having a greater temperature differential gives an increased conductance in this area.

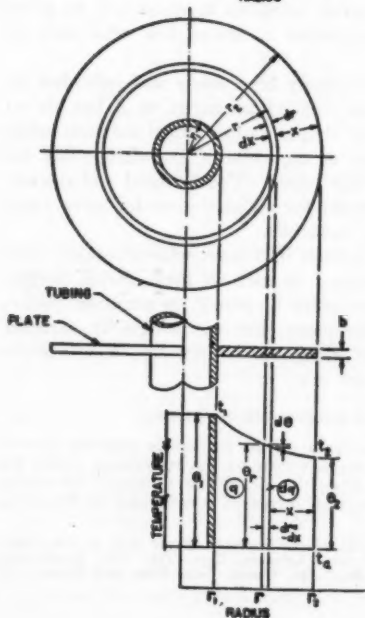
<sup>1</sup> S. R. Parsons and D. R. Harper, Nat. Bur. Standards Technol. Paper 211, p. 326, 1922; D. R. Harper and W. B. Brown, Nat. Adv. Comm. Aeronaut. Rept. 158, 1922, Government Printing Office, Washington, D. C.; E. Griffiths, Brit. Adv. Comm. Aero. Rept. and Memo. 308, 1917.



## LEGEND

- $b$  = Plate thickness, feet  
 $b_1$  = Plate thickness at  $r_1$   
 $b_2$  = Plate thickness at  $r_2$   
 $r$  = Radius, feet  
 $r_1$  = Inside radius of plate  
 $r_2$  = Outside radius of tubing  
 $r_3$  = Outside radius of plate  
 $dr$  = Differential radius  
 $x = r_2 - r$   
 $dx = -dr$   
 $t$  = Temperature, degrees Fahrenheit  
 $t_1$  = Surface temperature at  $r_1$   
 $t_2$  = Surface temperature at  $r_2$   
 $t_a$  = Temperature of ambient fluid  
 $t_m$  = Mean surface temperature of plate  
 $\theta$  = Temperature difference, degrees Fahrenheit  
 $\theta_1 = t_1 - t_a$   
 $\theta_2 = t_2 - t_a$   
 $\theta_r = t_r - t_a$   
 $\theta_m = t_m - t_a$   
 $d\theta$  = Differential temperature difference  
 $Q$  = Quantity of heat, Btu  
 $q$  = Rate of heat flow, Btu per hour  
 $dq$  = Differential rate of heat flow

FIG. 2. CONSTANT METAL CROSS-SECTION CIRCULAR PLATE HEATING FLUID DIAGRAM



## LEGEND

- $b$  = Plate thickness, feet  
 $r$  = Radius, feet  
 $r_1$  = Inside radius of plate  
 $r_2$  = Outside radius of tubing  
 $r_3$  = Outside radius of plate  
 $dr$  = Differential radius  
 $x = r_2 - r$   
 $dx = -dr$   
 $t$  = Temperature, degrees Fahrenheit  
 $t_1$  = Surface temperature at  $r_1$   
 $t_2$  = Surface temperature at  $r_2$   
 $t_a$  = Temperature of ambient fluid  
 $t_m$  = Mean surface temperature of plate  
 $\theta$  = Temperature difference, degrees Fahrenheit  
 $\theta_1 = t_1 - t_a$   
 $\theta_2 = t_2 - t_a$   
 $\theta_r = t_r - t_a$   
 $\theta_m = t_m - t_a$   
 $d\theta$  = Differential temperature difference  
 $Q$  = Quantity of heat, Btu  
 $q$  = Rate of heat flow, Btu per hour  
 $dq$  = Differential rate of heat flow

FIG. 3. FLAT CIRCULAR PLATE HEATING FLUID DIAGRAM

The average value, however, is probably not far out of line although the actual performance should be somewhat better than that calculated from the chart.

3. That the lines of heat flow are approximately radial.

4. That the error introduced by the assumption of logarithmic mean temperature over the surface of the fin for any single row of tubes in depth as determining the actual temperature difference is negligible. (This would be physically correct if it were not for the fact that the variation in air temperature distorts the line of heat flow and lengthens the path somewhat. Therefore, this assumption will give resistances somewhat below the actual.)

5. In plate fin construction, the assumption is made that the area served by each tube is equivalent in performance to a circular plate of equal area. The magnitude of the error introduced by this assumption either for a square or a rectangular spacing

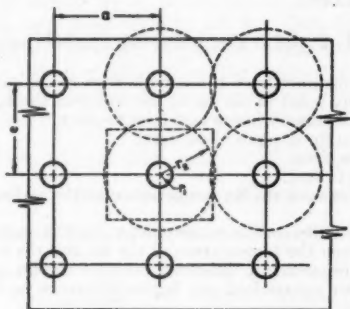


FIG. 4. SECTION OF THREE-ROW FLAT FIN COIL SHOWING EQUIVALENT CIRCULAR AND RECTANGULAR HEAT TRANSFER AREAS SERVED BY CIRCULAR TUBES

of tubes is discussed in Appendix IV, in which the approximate corrections to be applied are indicated.

The three main types of fins in commercial use are indicated in Figs. 1, 2 and 3. From the consideration of the geometry of each type of surface and the physical laws of heat transfer involved in conformance with the foregoing assumptions, a differential equation has been set up for each case and solved in the manner described in Appendices I, II and III. The first case, Fig. 1, corresponds to the solution for a *bar* fin. This fin may also be formed by winding a thin, narrow ribbon spirally about a tube. In this fin, the cross-section for transfer of heat is constant throughout the height of the fin and the perimeter is also constant throughout the height of the fin as the inner edge of the fin next to the tube is crimped but the metal is not appreciably stretched. In the second case, Fig. 2, a thicker ribbon is employed so that while the inner edge is not appreciably deformed, the outer edge is stretched and the metal thinned. In this fin, the cross-sectional area cut by a circle at any radius is constant but the perimeter or surface element exposed to air flow varies directly as the radius. In the third case, Fig. 3, which is the one primarily discussed in this paper, plates or discs of uniform thickness take the place of a spirally wound fin and the tubes pass perpendicularly

through these plates of uniform thickness. In the latter case, cross-sectional area for heat transfer varies directly as the radius and the perimeter or differential area subject to air flow also varies directly as the radius. This is the case of the plate fin.

The differential equations for the three typical constructions are as follows:

$$\frac{d^2\theta}{dx^2} = m^2\theta \quad \dots \dots \dots (1)$$

$$\frac{d^2\theta}{dx^2} = \frac{m^2\theta}{r_1} (r_2 - x) \quad \dots \dots \dots (2)$$

$$\frac{d^2\theta}{dx^2} - \left( \frac{1}{r_2 - x} \right) \frac{d\theta}{dx} = m^2\theta \quad \dots \dots \dots (3)$$

In the Equations 1, 2 and 3 the following symbols are employed:

$x = (r_2 - r)$  = the distance from the outside edge to the radius  $r$ , feet.

$r$  = the radius at any point in the fin to the center of the tube, feet.

$r_1$  = outside radius of tube = inside radius of fin, feet.

$r_2$  = outside radius of fin or equivalent, feet.

$L$  = height of the fin, feet.

$b$  = thickness of fin in feet.

$\theta$  = the difference between the fin temperature at the radius  $r$  and the ambient air temperature.

$\theta_1$  = the temperature difference between the fin and the ambient air at the radius  $r_1$ .

$\theta_2$  = difference between the temperature of the fin and the ambient air at radius  $r_2$ .

$\theta_m$  = effective mean temperature difference between the fin and ambient air.

$h$  = Btu per hour per square foot per degree difference = heat transfer coefficient from plate to air.

$k$  = Btu per hour per square foot per degree temperature difference per foot = conductivity of finned metal.

$E = \theta_m/\theta_1$  = fin effectiveness, dimensionless.

The final solution for each of these cases (see Appendix I, II and III) is as follows:

#### CASE I

$$\theta = \theta_2 \cdot \frac{1}{2} (e^{mx} + e^{-mx})$$

$$= \theta_1 \left[ \frac{e^{mx} + e^{-mx}}{e^{mL} + e^{-mL}} \right]$$

$$\frac{\theta_m}{\theta_1} = \frac{\tanh [m(r_2 - r_1)]}{m(r_2 - r_1)}$$

where

$$m = \sqrt{\frac{2h}{kb}}$$

#### CASE II

$$E = \frac{\theta_m}{\theta_1} = \frac{2R}{R+1} \left[ \frac{1 - \frac{(R-1)}{2R} + P^2 \left\{ (R-1)^2 \left( \frac{R+3}{24} \right) - \frac{(R-1)^2 (R+4)}{R} \left( \frac{R+4}{120} \right) \right\} + \dots}{1 + P^2 \left\{ \frac{R(R-1)^2}{2} - \frac{(R-1)^2}{6} \right\} + \dots} \right]$$

where

$$R = r_2/r_1$$

$$P = r_1 \sqrt{2h/kb_1}$$

$b_1$  = fin metal thickness at fin base

## CASE III

$$E = \frac{\theta_m}{\theta_1} = \left[ \frac{\frac{2R}{R+1} \left\{ 1 - \left( \frac{R-1}{2R} \right) + \frac{P^2}{6} (R-1)^2 - \frac{P^2}{12R} (R-1)^3 - \frac{(P^2 - P^4 R^2)}{120R^3} (R-1)^4 + \dots \right\}}{1 + \frac{P^2}{2} (R-1)^2 + \frac{P^2}{6R} (R-1)^3 + \left( \frac{P^2}{8R^2} + \frac{P^4}{24} \right) (R-1)^4 + \dots} \right]$$

where

$$R = r_2/r_1$$

$$P = r_1 \sqrt{2h/kb}$$

Numerical values for Case I are easily found. Case II and Case III, however, can only be solved by a power series as indicated and complete numerical values have only been worked out for Case III. These have been plotted in Fig. 5, which permits the mean temperature to be calculated immediately when the size and spacing of the tubes, the thickness of the fin and the coefficient of heat transfer on the air side are known. It will be noted that this chart is plotted in terms of dimensionless groups, which makes it of universal application. While the ratio of  $\frac{\theta_m}{\theta_1}$  is shown to be a function of  $h$ , the actual resistance of the fin to the flow of heat varies but slightly with  $h$  except for large values of  $P$  and  $\frac{r_2}{r_1}$ . This is discussed and the variation shown in Appendix V. For this reason, for most practical cases,  $R_m$  may be assumed to be a constant for different values of  $h$  in the formula

$$R_m = \left[ \frac{\theta_1}{\theta_m} - 1 \right] \frac{1}{h},$$

where  $h$  is given an average value of 12 Btu per hour per square foot per degree difference. A chart, Fig. 5, based on this assumption for Case III permits the resistance of the plate fin to be calculated with sufficient accuracy within the commercial limitations regardless of the variation in  $h$ .

A few illustrations of the practical use of these formulae and charts are desirable.

## COMPARISON OF PERFORMANCE OF TWO HEATERS BASED ON TEST RESULTS OF EACH

As mentioned in the forepart of the paper, the performance of two heaters can only be compared properly when they are tested or calculated for the same ratio of initial to final temperature difference and for the same pressure drop.<sup>2</sup> This relationship is developed in Appendix VI. The formula for this relationship is:

$$\frac{U_2}{U_1} = \left[ \frac{\Delta p_2 / \Delta p_1}{\log R_2 / \log R_1} \right]^{m/(1+n-m)}$$

<sup>2</sup> Report No. 731—A Proposed Method for Comparison of Effectiveness of Indirect Heating Surfaces, by A. E. Stacey, Jr., and C. M. Ashley. (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 257.)

where  $U_1$ ,  $p_1$ , and  $R_1$  are taken from a test point on an actual heater.  $\Delta p_2$  and  $R_2$  are assumed conditions to be met by the same heater with required change in the heater of the ratio of depth to face area and  $U_2$  is the corresponding heat transfer value under these conditions.  $m$  and  $n$  are air velocity

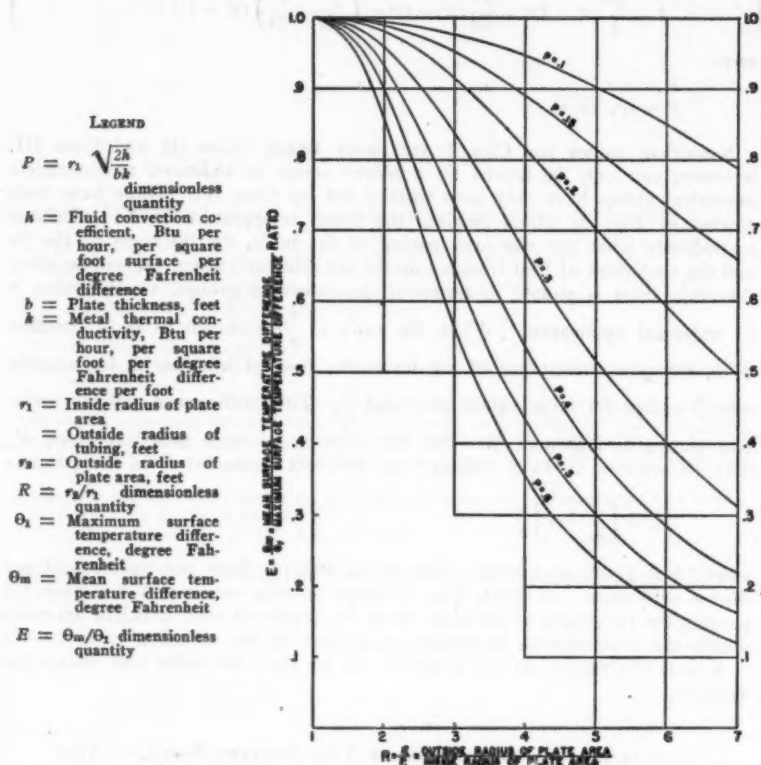


FIG. 5. CURVES SHOWING HEAT TRANSFER EFFECTIVENESS FROM A FLAT CIRCULAR OR EQUIVALENT RECTANGULAR PLATE ( $P$  vs.  $E$  AND  $R$ —RESIDUAL INCLUDED)

exponents expressing both  $u$  and  $\Delta p$ , respectively, in an exponential relationship with the heater face area velocity.

#### ILLUSTRATIONS OF PRACTICAL APPLICATION OF FORMULAE AND CHARTS

A few illustrations of the practical use of these formulae and charts is desirable.

Assume A a copper tube and fin and B a tube and fin of mild steel of the specifications given in the preceding tabulation. The relative results are also

given in the same tabulation. Tabulations for these examples appear in Tables 1 and 2.

It will be noted from this tabulation that the overall heat transfer is 8.83 for the copper and 7.92 for the steel. However, while the pressure drop for both heaters is the same, the air face velocity is different. The ratio of initial to final temperature difference is different for the two cases. Therefore, we will have to make a comparison applying the formula of Appendix VI for comparing two different heater surfaces. This is done in the preceding tabulation, in which we see that the relative value of the steel surface is approximately 89 per cent of that of the copper surface. If the cost of the steel surface as compared to the cost of the copper surface were approximately in the same ratio, there would be no choice between steel and copper commercially, aside from difference in resistance to corrosion. As a matter of fact, however, the steel surface actually costs somewhat more to produce than the copper surface, so there is no question when postwar restrictions are removed that copper will be used in preference to steel.

Let us now consider two examples under Case III, one A for a plate fin with square spacing of tubes and the other B for a rectangular spacing of tubes. On the basis of a comparison of the two performances, which are somewhat different, by the formula of Appendix VI it will be seen that the ratio of heat transfer under identical conditions of B is 99 per cent of that of A or, in other words, substantially the same, being an advantage of only 1 per cent of square arrangement over rectangular arrangement in the particular examples given. These examples are shown in Tables 3 and 4.

#### ACKNOWLEDGMENT

The authors wish to acknowledge the assistance of Robert B. Hyde in making original numerical computations for the flat plate performance chart, and to Byron W. Winborn, Jr., for suggesting the method of computing corrections for rectangular fins compared with equivalent circular fins of equal area, Appendix IV.

#### APPENDIX I

##### DERIVATION OF THE FORMULA FOR TRANSFER AND TEMPERATURE DROP THROUGH THE BAR FIN OR EQUIVALENT SPIRAL FINNED TUBE—CASE I

This is the simplest pattern of heat distribution in finned surfaces and has been amply covered by others.<sup>1</sup> The derivation of its mathematics is given here because of its relation to Cases II and III. The following symbols which refer to Fig. 1 are employed:

$L$  = height of the fin—feet.

$x$  = distance from outer edge of the fin to an element  $dx$ .

$dq = -h\delta dS$  = differential change in rate of heat flow through the laminar section  $dx \times bW$  which is located at the distance  $x$  from the outer edge of the fin.

$\delta$  = thickness of fin—feet.

$W = (2\pi r_2)$  = width of any fixed segment of the fin—feet.

$dS = -2Wdx$  = differential area from which heat is transferred to the ambient air—square feet.

$q$  = rate of heat flow through any laminar section of the fin at the distance  $x$  from the outer edge—Btu per hour.

$q_1$  = rate of heat flow through a section at the base of the fin.

$q_2 = 0$  (for convenience of calculation, it is assumed there is no heat conducted from the edge of the fin).

<sup>1</sup> S. R. Parsons and D. R. Harper, Nat. Bur. Standards Technol. Paper 211, p. 326, 1922.

TABLE 1—BAR FIN EQUIVALENT (CASE I)

EXAMPLE NO.	A	B
Surface material.....	Copper	Mild Steel
Fin spacing.....	8/inch	8/inch
Tube arrangement in adjacent rows.....	Staggered	Staggered
Rows deep.....	4	4
$a$ = Tube face spacing.....	0.116 ft	0.116 ft
$c$ = Tube row spacing.....	0.100 ft	0.100 ft
$b$ = Fin thickness.....	0.000833 ft	0.00125 ft
$r_2$ = Outside fin radius.....	0.0580 ft	0.0580 ft
$r_1$ = Fin base radius.....	0.0260 ft	0.0260 ft
$L$ = $r_2 - r_1$ = Fin height.....	0.0320 ft	0.0320 ft
$r_1'$ = Inside tube radius.....	0.0237 ft	0.0237 ft
$h_1$ = Fluid convection coefficient inside tube.....	600	600
$h_2$ = Air convection coefficient between fin and ambient air.....	12	12
$k$ = Fin metal thermal conductivity.....	220	30
$m = \sqrt{2h_2/kb}$ .....	11.45	25.3
$mL$ .....	0.366	0.810
$E = \frac{\tanh mL}{mL} = \frac{\theta_m}{\theta_1}$ .....	0.959	0.827
$S/A$ = Total exposed out. surf., sq ft surf./ (sq ft face area) (4 rows deep).....	81.8	81.4
$S_1/A$ = Inside tube surf., sq ft surf./ (sq ft face area) (4 rows deep).....	5.1	5.1
$S_p/A$ = Outside exposed tube surf., sq ft surf./ (sq ft face area) (4 rows deep).....	4.6	4.1
$S_t/A$ = Lateral fin surf., sq ft surf./ (sq ft face area) (4 rows deep).....	77.2	77.3
100 ( $A_1/A$ ) = Per cent minimum free face area.....	48.5	45.5
$R_m = \frac{1}{h_2} \left[ \frac{1-E}{E + \frac{S_p}{S_1}} \right]$ = Metal thermal resistance, (hr) (sq ft of $S$ ) (deg F)/Btu.....	0.0034	0.0164
$R_a = \frac{1}{h_2} S$ = Air film thermal resistance, (hr) (sq ft of (deg F)/Btu.....	0.0833	0.0833
$R_1 = \left( \frac{S}{S_1} \right) \frac{1}{h_1}$ = Inside tube film thermal resistance, (hr) (sq ft of $S$ ) (deg F)/Btu....	0.0265	0.0264
$R_s = R_a + R_m + R_1$ = Over-all thermal resistance, (hr) (sq ft of $S$ ) (deg F over-all)/Btu.....	0.1132	0.1261
$U = 1/R_s$ = Over-all heat transfer coefficient, Btu/(hr) (sq ft of $S$ ) (deg F over-all).....	8.83	7.92

TABLE 2—HEATER COMPARISON—BAR FIN EQUIVALENT (CASE I)  
(Copper vs. Steel Fin Heater Performance)

EXAMPLE NO.	A	B	BC
$V$ = Air face velocity, ft min.....	553	520	....
$S/A$ = Total outside surface, sq ft surf./ (sq ft face) (4 rows deep).....	81.8	81.4	....
$G$ = Air mass velocity, lb/(hr) (sq ft face).....	2490	2340	....
$U$ = Over-all heat transfer coefficient Btu/(hr) (sq ft of $S$ ) (deg F over-all).....	8.83	7.92	....
$C_p$ = Air specific heat, Btu/(lb) (deg F).....	0.24	0.24	....
$m$ = Air velocity heat transfer exponent.....	0.51	0.46	....
$n$ = Air velocity exponent of air friction.....	1.81	1.81	....
$\log R = \frac{U(S/A)}{C_p G} = \text{Log of temperature difference ratio},$ $(\theta_1/\theta_2)$ .....	1.21	1.15	1.21
$\Delta p$ = Air friction/4 rows deep (inches of water).....	0.51	0.51	0.51
$U_{BC} = U_B \left[ \frac{\Delta p_{BC}/\Delta p_B}{\log R_{BC}/\log R_B} \right]^{(m)/(1+n-m)}$ .....	....	....	7.83
$U_{BC}/U_A = \frac{7.83}{8.83}$ .....	....	....	0.887

TABLE 3—FLAT, PLATE FIN (CASE III)

EXAMPLE NO.	A	B
Surface material.....	Copper	Copper
Tube arrangement in adjacent rows.....	<i>In Line</i>	<i>In Line</i>
Fin spacing.....	6/in.	6/in.
Rows deep.....	4	4
$a$ = Tube face spacing = Fin length/tube.....	0.167 ft	0.208 ft
$c$ = Tube row spacing = Fin depth/tube.....	0.167 ft	0.125 ft
$b$ = Fin thickness.....	0.000833 ft	0.000833 ft
$r_2$ = Radius of equivalent circular area = $\sqrt{\frac{ac}{\pi}}$ .....	0.0940 ft	0.0911 ft
$r_1$ = Fin base radius.....	0.0260 ft	0.0260 ft
$r_1'$ = Inside tube radius.....	0.0237 ft	0.0237 ft
$h_1$ = Fluid convection coefficient inside tube.....	600	600
$h_a$ = Air convection coefficient between fin and ambient air.....	10	10
$k$ = Fin metal thermal conductivity.....	220	220
$r_2/r_1$ = See Appendix IV, <i>Applications</i> .....	3.61	3.51
$P$ = See Appendix IV, <i>Applications</i> .....	0.272	0.272
$\bar{E} = \theta_m/\theta_1$ See Appendix IV, <i>Applications</i> .....	0.768	0.772
$S/A$ = Total exposed outside surf., sq ft surf./ (sq ft face area) (4 rows deep).....	92.5	69.0
$S_1/A$ = Inside tube surf., sq ft surf./ (sq ft face area) (4 rows deep).....	3.7	2.9
$S_p/A$ = Outside exposed tube surf., sq ft surf./ (sq ft face area) (4 rows deep).....	3.7	2.9
$S_l/A$ = Lateral fin surf., sq ft surf./ (sq ft face area) (4 rows deep).....	88.8	66.1
100 ( $A_1/A$ ) = Per cent minimum free face area.....	64.5	70.5
$R_m = \frac{1}{h_a} \left[ \frac{1 - \bar{E}}{\bar{E} + \frac{S_p}{S_f}} \right]$ = Metal thermal resistance, (hr) (sq ft of $S$ ) (deg F)/ Btu.....	0.0287	0.0280
$R_a = 1/h_a$ = Air film thermal resistance, (hr) (sq ft of $S$ ) (deg F)/Btu.....	0.1000	0.100
$R_1 = \left( \frac{S}{S_1} \right) 1/h_1$ = Inside tube film thermal resistance, (hr) (sq ft of $S$ ) (deg F)/Btu.....	0.0430	0.040
$R_s = R_a + R_m + R_1$ = Over-all thermal resistance, (hr) (sq ft of $S$ ) (deg F over-all)/Btu.....	0.1717	0.168
$U = 1/R_s$ = Over-all heat transfer coefficient, Btu/(hr) (sq ft of $S$ ) (deg F over-all).....	5.82	5.95

TABLE 4—HEATER COMPARISON—FLAT, PLATE FIN (CASE III)

(Square vs. Rectangular Plate Fin Heater Performance)

EXAMPLE NO.	A	B	BC
$V$ = Air face velocity, ft/min.....	1250	1370	....
$S/A$ = Total outside surface, sq ft surf./ (sq ft face) (4 rows deep).....	92.5	69.0	....
$G$ = Air mass velocity, lb/(hr) (sq ft face).....	5630	6170	....
$U$ = Over-all heat transfer coefficient Btu/ (hr) (sq ft of $S$ ) (deg F over-all).....	5.82	5.95	....
$C_p$ = Air specific heat, Btu/(lb) (deg F).....	0.24	0.24	....
$m$ = Air velocity heat transfer exponent.....	0.38	0.38	....
$n$ = Air velocity exponent of air friction.....	1.71	1.71	....
$\log R = \frac{U(S/A)}{C_p G}$ = Log of temperature difference ratio, $(\theta_1/\theta_2)$ .....	0.398	0.277	0.398
$\Delta p$ = Air friction/4 rows deep (inches of water).....	0.54	0.45	0.54
$U_{BC} = U_B \left[ \frac{\Delta p_{BC}/\Delta p_B}{\log R_{BC}/\log R_B} \right]^{(m/1+n-m)}$ .....	....	....	5.78
$U_{BC}/U_A = \frac{5.78}{5.82}$ .....	....	....	0.99

$\theta$  = temperature difference between the ambient air and the fin at the distance  $x$  from the outer edge of the fin.

$\theta_1$  = temperature difference between the air and the base of the fin.

$\theta_2$  = temperature difference between the ambient air and the outer edge of the fin.

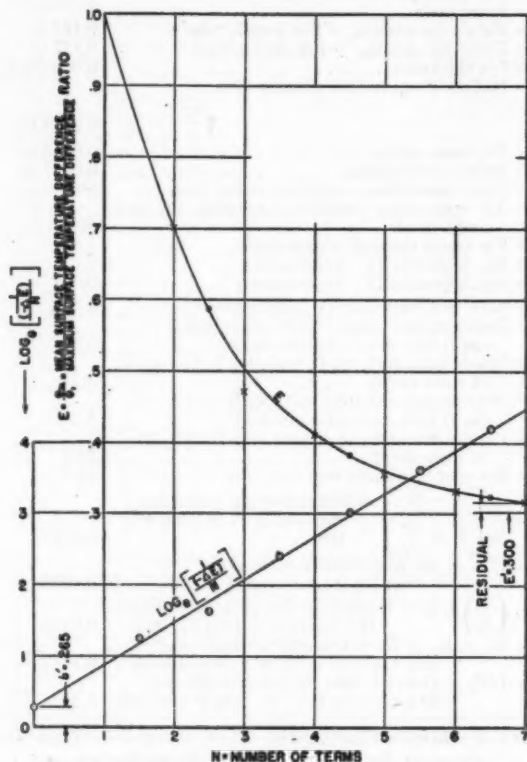


FIG. 6. CURVES SHOWING VARIATION OF HEAT TRANSFER EFFECTIVENESS ( $E$  VALUE) WITH NUMBER OF TERMS ( $N$ ) FOR POINT WHERE  $P=0.5$  AND  $R=5$  (APPENDIX III)

Method of determining correction for residual error between seven terms and an infinite number of terms for Case III

$h$  = coefficient of heat transfer from a surface to the air.

$k$  = thermal conductivity of the fin metal.

The law of temperature change in this surface (or any other) may be expressed by a differential equation formed by equating two expressions for change in heat flow; *vis*, (1) the rate of heat flow from the differential area  $Wdx$  to the air stream,  $dq = h\theta(2Wdx)$ , and (2) the rate of heat flow through the cross sectional area  $Wb$  as derived from the resistance formula  $\frac{d\theta}{dx} = \frac{q}{kWb}$ .

Then:

$$\frac{dq}{dx} = 2h\theta W \text{ from the first} \quad (1)$$

$$\frac{d\theta}{dx} = \frac{q}{kWb} \text{ from the second} \quad (2)$$

Taking the derivative of both sides of Equation (2) with respect to  $x$ , we have

$$\frac{d^2\theta}{dx^2} = \frac{1}{kWb} \frac{dq}{dx} \text{ or} \quad (3)$$

$$\frac{dq}{dx} = kWb \frac{d^2\theta}{dx^2} \quad (4)$$

Combining Equations (1) and (4),

$$\frac{d^2\theta}{dx^2} = \left(\frac{2h}{kb}\right)\theta \text{ or } \frac{d}{dx}\left(\frac{d\theta}{dx}\right) = \left(\frac{2h}{kb}\right)\theta \quad (5)$$

This basic differential equation may be solved very simply by ordinary integration as follows: Multiply both sides of Equation (5) by  $\left(\frac{d\theta}{dx}\right)$  Then

$$\frac{d\theta}{dx} \cdot \frac{d}{dx}\left(\frac{d\theta}{dx}\right) = \left(\frac{2h}{kb}\right)\theta \frac{d\theta}{dx} \quad (6)$$

This is equivalent to

$$\begin{aligned} \frac{1}{2} \frac{d}{dx}\left(\frac{d\theta}{dx}\right)^2 &= \left(\frac{2h}{kb}\right)\theta \frac{d\theta}{dx} \text{ or} \\ \frac{1}{2} d\left(\frac{d\theta}{dx}\right)^2 &= f(x)dx = \left(\frac{2h}{kb}\right)\theta d\theta \end{aligned} \quad (7)$$

Integrating the first term of Equation (7) between the limits  $x = 0$  and  $x = x$  and corresponding limits  $\frac{d\theta}{dx} = 0^4$  and  $\frac{d\theta}{dx} = \frac{d\theta}{dx}$ , and integrating the second member of the second equation between the corresponding limits  $\theta_1$  and  $\theta$ , we have

$$\begin{aligned} \frac{1}{2} \left(\frac{d\theta}{dx}\right)^2 - \left(\frac{1}{2} \times 0\right) &= \frac{1}{2} \left(\frac{2h}{kb}\right) [\theta^2 - \theta_1^2] \text{ or } \frac{d\theta}{dx} = \sqrt{\frac{2h}{kb}} \sqrt{\theta^2 - \theta_1^2} \text{ or} \\ \sqrt{\frac{2h}{kb}} dx &= \frac{d\theta}{\sqrt{\theta^2 - \theta_1^2}} \end{aligned} \quad (8)$$

Then

$$\begin{aligned} \sqrt{\frac{2h}{kb}} \int_0^x dx &= \int_{\theta_1}^{\theta} \frac{d\theta}{\sqrt{\theta^2 - \theta_1^2}} \text{ or} \\ x \sqrt{\frac{2h}{kb}} &= \log \left[ \frac{\theta + \sqrt{\theta^2 - \theta_1^2}}{\theta_1 + \sqrt{\theta_1^2 - \theta_1^2}} \right] \\ &= \log \left[ \left(\frac{\theta}{\theta_1}\right) + \sqrt{\left(\frac{\theta}{\theta_1}\right)^2 - 1} \right] = \cosh^{-1} \left(\frac{\theta}{\theta_1}\right) \end{aligned} \quad (9)$$

This will give the temperature at any point on the fin if  $\theta_1$  is known, including the point  $x = L = (r_2 - r_1)$ ; i.e., where  $\theta$  equals  $\theta_1$  which is known. Therefore,

<sup>4</sup> Since  $q = 0$  when  $x = 0$  and therefore

$$d \left\{ \frac{d\theta}{dx} \right\} = \frac{q}{kWb} = 0 \text{ when } x = 0.$$

$$L \sqrt{\frac{2h}{kb}} = \log \left[ \frac{\theta_1}{\theta_2} + \sqrt{\left( \frac{\theta_1}{\theta_2} \right)^2 - 1} \right] \quad (10)$$

From Equation (10) and putting  $\sqrt{\frac{2h}{kb}} = m$  for convenience

$$\frac{\theta_1}{\theta_2} = \frac{1}{2} [e^{mL} + e^{-mL}] = \cosh [L \sqrt{\frac{2h}{kb}}]$$

$$\theta_2 = \frac{\theta_1}{\frac{1}{2} (e^{mL} + e^{-mL})} \quad (11)$$

From Equations (9) and (11)

$$\theta = \theta_2 \cdot \frac{1}{2} (e^{mx} + e^{-mx}) = \theta_1 \left[ \frac{e^{mx} + e^{-mx}}{e^{mL} + e^{-mL}} \right] \quad (12)$$

One is now able to integrate the rate of heat transfer to the air over the entire surface and also to determine the mean temperature difference,  $\theta_m$ . Then

$$\int_0^{S_2} dq = \int_0^{S_2} h \theta dS = \int_0^L 2hW \theta dx = \int_{r_1}^{r_2} 2hW \theta dr$$

See Equation (1).

Introducing the value of  $\theta$  from Equation (12),

$$\int_0^{S_2} dq = 2hW \theta_1 \int_0^L \frac{1}{2} [e^{mx} + e^{-mx}] dx \quad (13)$$

$$\text{Since } \int e^{mx} dx = \frac{1}{m} \int e^{mx} d(mx) = \frac{1}{m} e^{mx}$$

$$q_s = \frac{2hW \theta_1}{m} \cdot \frac{1}{2} [(e^{mL} - 1) - (e^{-mL} - 1)]$$

$$= \frac{2hW \theta_1}{m} \left[ \frac{e^{mL} - e^{-mL}}{e^{mL} + e^{-mL}} \right] = 2hWL \theta_1 \left[ \frac{\tanh mL}{mL} \right] \quad (14)$$

noting that  $mL$  is a dimensionless number.

$$q_s = h \theta_m S_2 \text{ where } S_2 = 2WL. \text{ Therefore,}$$

$$\theta_m = \frac{q_s}{2hWL} = \frac{\theta_1}{mL} \left[ \frac{e^{mL} - e^{-mL}}{e^{mL} + e^{-mL}} \right] \quad (15)$$

or

$$\frac{\theta_m}{\theta_1} = \frac{\tanh mL}{mL} \text{ where } m = \sqrt{\frac{2h}{kb}} \quad (15a)$$

or

$$\frac{\theta_m}{\theta_1} = \frac{\tanh m(r_2 - r_1)}{m(r_2 - r_1)} \quad (15b)$$

It will be seen that Equations (14) and (15) give the practical solutions required and a chart in terms of the dimensionless groups,  $\frac{\theta_m}{\theta_1}$ ,  $\frac{r_2}{r_1}$  and  $P = r_1 \sqrt{\frac{2h}{kb}}$  could be plotted as in Fig. 5, but this is not necessary as in Cases II and III because of the simplicity of the Case I solution.

#### RESISTANCE

The only remaining equation required is that giving the thermal resistance to heat flow through the fin in order to determine the overall conductance,  $U = \frac{1}{R_s}$ , where  $R_s$  = the summation of three principal resistances.

$$R_s = R_t + R_m + R_a = \frac{1}{h_t} + \frac{1}{C_m} + \frac{1}{h_a} \text{ where}$$

$R_t$  = the film resistance of the heating fluid on the inside of the tube referred to surface (S)<sup>3</sup>

$R_m$  = the metal resistance of the fin

$R_a$  = the resistance of the air film

and  $h_t$ ,  $C_m$ ,  $h_a$  are the respective conductances referred to surface (S)<sup>3</sup>.

In heat transfer under the steady state, the equation

$$q/S = C_m(t_i - t_m) \text{ i.e. } C_m = \frac{d(q/S)}{d\theta}$$

$$\frac{(\theta_i - \theta_m)}{R_m} = q/S = h\theta_m \text{ where } h \text{ is the coefficient of heat transfer from surface to air} \quad (16)$$

$$R_m = \frac{\theta_i - \theta_m}{h\theta_m} = \frac{1}{h} \left[ \left( \frac{\theta_i}{\theta_m} \right) - 1 \right]$$

$$= \frac{1}{h} \left[ \frac{1 - \left( \frac{\theta_m}{\theta_i} \right)}{\left( \frac{\theta_m}{\theta_i} \right)} \right] \quad (17)$$

This relationship obviously applies equally to Cases I, II and III or any others. In Case I,

$$R_m = \frac{1}{h} \left[ \frac{mL}{\tanh mL} - 1 \right] \text{ where } m = \sqrt{\frac{2h}{kb}} \quad (18)$$

Apparently this resistance is not constant, as one might suppose, but is an involved function of  $h$ . However, in most practical designs of finned tubing, the effect of the variation of  $h$  through considerable range is nearly negligible, so that the fin resistance  $R_m$ , calculated for an average value of  $h$  is satisfactory for other values of  $h$ . That is, it is not important that we know the actual value of  $h$  in order to calculate the resistance of the fin.

## APPENDIX II

### DERIVATION OF FORMULA FOR THE TEMPERATURE CHANGE THROUGH A CIRCULAR FIN OF CONSTANT METAL CROSS SECTION—CASE II

Symbols:

As in Appendix I with the following exceptions:

$b_1$  = fin metal thickness at fin base, feet, since  $b$  is variable for this case.

$$R = \frac{r_2}{r_1}, \text{ dimensionless}$$

$$m = \sqrt{\frac{2h}{kb_1}}, \frac{1}{\text{ft}}$$

$$P = r_1 m = r_1 \sqrt{\frac{2h}{kb_1}}, \text{ dimensionless}$$

$$E = \frac{\theta_m}{\theta_i} = \text{fin effectiveness, dimensionless}$$

<sup>3</sup> All thermal resistances must be referred to the same surface. For example, the effective inside film resistance ( $R_t$ ) referred to the total outside surface (S) is equal to the specific inside film resistance multiplied by the ratio of total outside surface to inside surface.

## DEVELOPMENT OF FORMULA

The differential equation, expressing the temperature change through a fin of uniform metal cross section such as shown in Fig. 2, is determined in a manner similar to the detailed derivation procedure outlined in Appendix I.

The constant fin metal cross section requires that

$$\text{If } r_1 b_1 = r_2 b_2 = r b$$

$$m = \sqrt{\frac{2h}{kb_1}}$$

and heat transmission through the outside fin edge area ( $2\pi r_2 b_2$ ) be neglected, the following differential equation is obtained:

$$\frac{d^2\theta}{dx^2} = \frac{m^2\theta(r_2 - x)}{r_1} \quad (1)$$

Equation (1), in Appendix II, is that of Case II in the text, and may be solved by expressing  $\theta$  in terms of a power series of  $x$ . The procedure outlined below is employed to develop an expression for the fin effectiveness,  $E$ .

The boundary conditions, from an inspection of Fig. 2, are as follows:

$$\theta = \theta_1, \text{ when } x = 0 \quad (a)$$

$$\frac{d\theta}{dx} = 0, \text{ when } x = 0 \text{ (neglecting outside fin edge heat transmission).} \quad (b)$$

$$\theta = \theta_1, \text{ when } x = (r_2 - r_1) \quad (c)$$

Let

$$\theta = A + Bx + Cx^2 + Dx^3 + \quad (2)$$

The first derivative of Equation (2) is

$$\frac{d\theta}{dx} = B + 2Cx + 3Dx^2 + \quad (3)$$

and the second derivative is

$$\frac{d^2\theta}{dx^2} = 2C + 6Dx + \quad (4)$$

Substitution of boundary condition (a) in Equation (2) determines the coefficient  $A$ , thus

$$A = \theta_1.$$

Substituting boundary condition (b) in Equation (3) establishes the coefficient  $B$ , thus

$$B = 0.$$

The remaining undetermined coefficients are evaluated by substituting the values of  $\theta$  and  $\frac{d^2\theta}{dx^2}$  from Equations (2) and (4), respectively, in Equation (1), hence

$$2C + 6Dx + \dots = \frac{m^2}{r_1}$$

$$[Ar_2 + (Br_2 - A)x + (Cr_2 - B)x^2 + \dots] \quad (5)$$

Solving for coefficients in terms of the previously established values of  $A$  and  $B$ , by equating coefficients of the like exponents of  $x$  in Equation (5), we obtain

$$C = \frac{\theta_1 m^2 r_2}{2r_1}$$

$$D = -\frac{\theta_1 m^2}{6r_1} \dots \dots \text{etc.}$$

In a similar manner, any number of coefficients for the higher powers of  $x$  are evaluated.

Substituting the determined coefficients in Equation (2), we find

$$\theta = \theta_2 \left[ 1 + \left( \frac{m^2 r_2}{2r_1} \right) x^2 - \left( \frac{m^4}{6r_1} \right) x^3 + \dots \right] \quad (6)$$

The total heat transmitted from the lateral fin surface to the ambient fluid, neglecting the outside fin edge, may be expressed as follows:

$$q = 2\pi(r_2^2 - r_1^2)h\theta_m = 4\pi h \int_0^{r_2-r_1} \theta(r_2 - x) dx \quad (7)$$

Then from Equation (7),

$$\theta_m = \frac{2}{r_2^2 - r_1^2} \int_0^{r_2-r_1} \theta(r_2 - x) dx \quad (7a)$$

Substituting the value of  $\theta$  from Equation (6) in (7a)

$$\begin{aligned} \frac{\theta_m}{\theta_2} = \frac{2}{r_2^2 - r_1^2} \int_0^{r_2-r_1} \left[ r_2 - x + \frac{m^2 r_2}{2r_1} (r_2 x^2 - x^3) - \right. \\ \left. \frac{m^4}{6r_1} (r_2 x^3 - x^4) + \dots \right] dx \quad (8) \end{aligned}$$

Integrating Equation (8), and substituting limits

$$\begin{aligned} \frac{\theta_m}{\theta_2} = \frac{2}{r_2 + r_1} \left[ (r_2 - \frac{r_2^2 - r_1^2}{2}) + \frac{m^2 r_2}{2r_1} \left\{ \frac{r_2(r_2 - r_1)^2}{3} - \frac{(r_2 - r_1)^3}{4} \right\} - \frac{m^4}{6r_1} \right. \\ \left. \left\{ \frac{r_2(r_2 - r_1)^3}{4} - \frac{(r_2 - r_1)^4}{5} \right\} + \dots \right] \quad (9) \end{aligned}$$

Let

$$r_2 = \frac{PR}{m} \quad (10)$$

and

$$r_1 = \frac{P}{m} \quad (11)$$

where

$$P = r_1 \sqrt{\frac{2h}{kb_1}}$$

Substituting the values of  $r_2$  and  $r_1$  from Equations (10) and (11), respectively, in Equation (9)

$$\begin{aligned} \frac{\theta_m}{\theta_2} = \frac{2R}{R+1} \left[ 1 - \frac{(R-1)}{2R} + P^2 \left\{ (R-1)^2 \left( \frac{R+3}{24} \right) - \frac{(R-1)^3}{R} \left( \frac{R+4}{120} \right) \right\} + \dots \right] \quad (12) \end{aligned}$$

Substituting boundary condition (c) and the values of  $r_2$  and  $r_1$  from Equations (10) and (11), respectively, in Equation (6), we find the following relation

$$\frac{\theta_1}{\theta_2} = \left[ 1 + P^2 \left\{ \frac{R(R-1)^2}{2} - \frac{(R-1)^3}{6} \right\} + \dots \right] \quad (13)$$

The total heat transmission ( $q$ ), from the lateral fin surface ( $S_l$ ), is expressed as

$$q = hS_l\theta_m \quad (14)$$

or

$$q = hS_lE\theta_1 \quad (14a)$$

Equating the values of ( $q$ ) from Equations (14) and (14a), the value of ( $E$ ) in terms of the temperature difference ratio is

$$E = \frac{\theta_m}{\theta_1} \quad (15)$$

From Equation (15) it is obvious that the fin effectiveness ( $E$ ), numerically equals that fraction of the lateral fin surface ( $S_l$ ), which, if operated at the prime or tube surface temperature difference ( $\theta_1$ ), would dissipate an amount of heat equaling the fin's total heat transmission ( $q$ ).

Finally, the formula for the fin effectiveness ( $E$ ), expressed as a function of the two dimensionless parameters,  $R$  and  $P$ , is obtained by dividing Equation (12) by (13), hence

$$E = \frac{\theta_m}{\theta_1} = \frac{2R}{R+1} \left[ \frac{1 + f_1(R) + P^2 f_2(R) + \dots}{1 + P^2 f_3(R) + \dots} \right] \quad (16)$$

where

$$f_1(R) = -\left(\frac{R-1}{2R}\right)$$

$$f_2(R) = \frac{(R-1)^2}{24} \left[ R+3 - \frac{(R-1)(R+4)}{5R} \right]$$

$$f_3(R) = (R-1)^2 \left[ \frac{2R+1}{6} \right]$$

### APPENDIX III

#### DERIVATION OF FORMULA FOR THE TEMPERATURE CHANGE THROUGH A CIRCULAR FIN PLATE OF UNIFORM THICKNESS—CASE III

*Symbols*

The same as in Appendix I with the following exceptions:

$$R = \frac{r_2}{r_1}, \text{ dimensionless}$$

$$P = r_1 m = r_1 \sqrt{\frac{2h}{kb}}, \text{ dimensionless}$$

$$E = \frac{\theta_m}{\theta_1} = \text{fin effectiveness, dimensionless}$$

#### DEVELOPMENT OF FORMULA

The differential equation, expressing the temperature change through a circular fin of uniform thickness ( $b$ ), such as illustrated in Fig. 3, is found in a manner similar to the detailed derivation procedure outlined in Appendix I.

If

$$m = \sqrt{\frac{2h}{kb}}$$

and heat transmission through the outside fin edge area ( $2\pi r_2 b$ ) be neglected, the following differential equation is obtained:

$$\frac{d^2\theta}{dx^2} - \left(\frac{1}{r_2 - x}\right) \frac{d\theta}{dx} = m^2\theta \quad (1)$$

Equation (1), above, is that of Case III in the text, and may be solved by expressing  $\theta$  in terms of a power series of  $x$ . The forthcoming procedure is used to develop an expression for the fin effectiveness ( $E$ ).

From an inspection of Fig. 3, the boundary conditions are established as follows:

$$\theta = \theta_1, \text{ when } x = 0 \quad \dots \dots \dots (a)$$

$$\frac{d\theta}{dx} = 0, \text{ when } x = 0 \text{ (neglecting outside fin edge heat transmission)} \quad \dots (b)$$

$$\theta = \theta_1, \text{ when } x = (r_2 - r_1) \quad \dots \dots \dots (c)$$

Let

$$\theta = A + Bx + Cx^2 + Dx^3 + Ex^4 + \dots \dots \dots (2)$$

Then by identical detailed procedure as that in Appendix II, the coefficients of Equation (2) are determined, and yield the following values:

$$A = \theta_1$$

$$B = 0$$

$$C = \frac{\theta_2 m^2}{2}$$

$$D = \frac{\theta_2 m^3}{6r_2}$$

$$E = \theta_2 \left( \frac{m^3}{8r_2^2} + \frac{m^4}{24} \right) \dots \dots \text{etc.}$$

Substituting the preceding values of the determined coefficients in Equation (2).

$$\theta = \theta_1 \left[ 1 + \left( \frac{m^2}{2} \right) x^2 + \left( \frac{m^3}{6r_2} \right) x^3 + \left( \frac{m^3}{8r_2^2} + \frac{m^4}{24} \right) x^4 + \dots \dots \right] \quad \dots \dots (3)$$

Substituting

$$r_2 = \frac{PR}{m}$$

$$r_1 = \frac{P}{m}$$

and

$$x = r_2 - r_1 [\text{Boundary condition (c)}] \text{ in Equation } \dots \dots \dots (3)$$

$$\frac{\theta_1}{\theta_2} = 1 + \frac{P^2}{2} (R-1)^2 + \frac{P^3}{6R} (R-1)^3 + \left( \frac{P^3}{8R^2} + \frac{P^4}{24} \right) (R-1)^4 + \dots \dots (4)$$

The value of  $\frac{\theta_m}{\theta_1}$  is found by the same method outlined in Appendix II and we thus determine the following relation:

$$\begin{aligned} \frac{\theta_m}{\theta_1} = & \frac{2R}{R+1} \left[ 1 - \left( \frac{R-1}{2R} \right) + \frac{P^2}{6} (R-1)^2 \right. \\ & \left. - \frac{P^3}{12R} (R-1)^3 - \frac{(P^3 - P^4 R^2)}{120R^2} (R-1)^4 + \dots \dots \right] \quad \dots \dots (5) \end{aligned}$$

The final expression for fin effectiveness ( $E$ ) is that obtained by dividing Equation (5) by Equation (4), thus

$$E = \frac{\theta_m}{\theta_1} = \left[ \frac{\frac{2R}{R+1} \left\{ 1 - \left( \frac{R-1}{2R} \right) + \frac{P^2}{6} (R-1)^2 - \frac{P^3}{12R} (R-1)^3 - \frac{(P^3 - P^4 R^2)}{120R^2} (R-1)^4 + \dots \right\}}{1 + \frac{P^2}{2} (R-1)^2 + \frac{P^3}{6R} (R-1)^3 + \left( \frac{P^3}{8R^2} + \frac{P^4}{24} \right) (R-1)^4 + \dots} \right] \quad (6)$$

Data for Fig. 5 were computed from Equation (6), in which the value of  $E$  was determined by expanding the power series of  $P$  and  $R$  to seven terms. With increasing values of either  $P$  or  $R$ , the series converges less rapidly.

Without going into detail, this question of series convergency was investigated and a method obtained to correct the value of  $E$  for an infinite number of terms in the power series expression. Each point calculated for preparation of Fig. 5 was corrected for the residual error in  $E$  before final curve plotting. Fig. 6 demonstrates the method employed to correct  $E$  for its residual error at given values of both  $P$  and  $R$ .

## APPENDIX IV

### A METHOD OF APPROXIMATING THE ERROR IN FIN EFFECTIVENESS ( $E$ ) WHEN CIRCULAR PLATE FIN DATA ARE APPLIED TO A RECTANGULAR OR SQUARE PLATE FIN

#### *Symbols:*

$S$  = Fin plate lateral surface area, square feet.

$a$  = Fin plate length/tube, feet.

$b$  = Fin plate thickness, feet.

$c$  = Fin plate width/tube, feet.

$E = \frac{\theta_m}{\theta_1}$  = Fin plate effectiveness, dimensionless.

$h$  = Ambient fluid convection coefficient, Btu/(hr) (sq ft) (deg F).

$k$  = Fin metal thermal conductivity, Btu/(hr) (sq ft) (deg F/ft).

$m = a/N$  = Fin plate length segment, feet.

$n = c/N$  = Fin plate width segment, feet.

$N$  = Total number of sectors, dimensionless.

$P = r_1 \sqrt{\frac{2h}{kb}}$ , dimensionless.

$q$  = Heat flow rate, Btu/hr.

$r$  = Sector radius, feet.

$r_1$  = Inside sector radius = outside tube radius, feet.

$r_2$  = Outside sector radius, feet.

$R = r_2/r_1$ , dimensionless.

$T = a/c$ , dimensionless.

$\theta_1$  = Temperature difference between fin plate at radius ( $r_1$ ) and ambient fluid, deg F.

$\theta_m$  = Mean temperature difference between fin plate and ambient fluid, degree Fahrenheit.

#### METHOD

A rectangular or square plate fin of uniform thickness is a commercial type of heat transfer surface used extensively. Its effectiveness ( $E$ ) is closely approximated by applying data of the flat, circular plate fin. It is the purpose of this discussion to indicate the magnitude of this error in the fin effectiveness ( $E$ ), based on the assumption of a flat circular fin, equivalent in area to the rectangular or square fin.

1. *Estimation of Fin Effectiveness ( $E$ ) by Equivalent Circular Area:* A reasonably close approximation of  $E$  for a rectangular or square plate is obtained by this simple method. The amount of error contained in  $E$ , thus computed, increases with an increment of either of its two dimensionless parameters,  $P$  or  $R$ .

Consider a rectangular fin plate of length ( $a$ ), width ( $c$ ) and uniform thickness ( $b$ ) with a tube of outside radius ( $r_1$ ) located at the fin's geometrical center, such as shown in Fig. 4. The circular fin, equivalent in area to the rectangular plate section, is shown in Fig. 4 as that of radius ( $r_2$ ). The value of  $r_2$  is computed thus:

$$r_2 = \sqrt{\frac{ac}{\pi}}$$

$$\text{Then } R = \frac{r_1}{r_2}$$

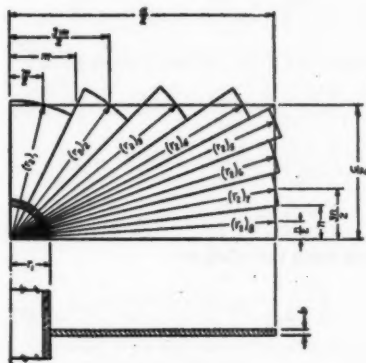
Also any value of the fluid convection coefficient ( $h$ ) for a given fin plate and tube allows the computation of

$$P = r_1 \sqrt{\frac{2h}{kb}}$$

With values of  $R$  and  $P$ , thus established, the mean fin effectiveness,  $\bar{E}$ , is read directly from data of the circular plate fin as shown in Fig. 5. The heat transferred through the total fin surface ( $S$ ) is then:

$$q = hS\bar{E}\theta_1$$

2. *Estimation of Fin Effectiveness ( $E$ ) by Segmentation of Fin Plate:* This procedure yields a closer estimate of  $E$  than that by equivalent circular area method. Referring to Fig. 4, if the fin plate is divided into four equal sections, formed by passing two mutually



#### LEGEND

- $a$  = Fin plate length per tube, feet
- $b$  = Fin plate thickness, feet
- $c$  = Fin plate width per tube, feet
- $r_s$  = Outside sector radius, feet
- $r_1$  = Outside tube radius, feet
- $N$  = Total number of sectors
- $m = a/N$  = fin plate length segment, feet
- $n = c/N$  = fin plate width segment, feet

FIG. 7. RECTANGULAR PLATE FIN QUADRANT SHOWING SEGMENTATION (APPENDIX IV)

perpendicular planes through the tube center such that the fin length ( $a$ ) and fin width ( $c$ ) are each bisected, then the fin temperature of a corresponding point in all quadrants is identical, at least based on the assumptions made for the derivation of  $E$ . Hence, analysis of a single quadrant is the only requirement.

Heat flow through the fin is non-radial since the varying heat path lengths around the fin perimeter produce a circumferential component of the fin temperature gradient. If the fin quadrant is sub-divided into a number of sectors bounded by radii from the tube center, then the error in assuming radial heat flow in any sector is reduced. Obviously, this error decreases with an increase in the number of sectors selected.

Consider a fin quadrant of length  $\left(\frac{a}{2}\right)$ , width  $\left(\frac{c}{2}\right)$  and fin thickness ( $b$ ). Let the quadrant be sub-divided into  $N$  sections such that  $\frac{N}{2}$  sections are formed from equal segments of both its length and width. Then, depending on the value of  $N$ , each section may be approximately represented by a sector of radius ( $r_s$ ). In the forthcoming method the sectors are assumed as actually representing the fin plate, since, with the number of sectors chosen, the sector areas very nearly represent the actual section areas.

Radial heat flow is assumed in all sectors. This assumption is only exact if the fin were slitted between sectors.

Fig. 7 illustrates a typical quadrant arbitrarily subdivided into eight sectors ( $N = 8$ ). The sector radii ( $r_s$ ) are computed as follows:

$$(r_s)_1 = \sqrt{\left(\frac{c}{2}\right)^2 + \left(\frac{m}{2}\right)^2} \dots \dots \dots (1)$$

$$\text{But } c = \frac{a}{T} \dots \dots \dots (2)$$

$$\text{And } m = \frac{a}{N} \dots \dots \dots (3)$$

Substituting values of  $c$  and  $m$  from Equations (2) and (3), respectively, in Equation (1) and simplifying, we find

$$(r_s)_1 = \frac{a}{2} \sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{1}{N}\right)^2} \dots \dots \dots (4)$$

Similarly,

$$(r_s)_2 = \frac{a}{2} \sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{3}{N}\right)^2} \dots \dots \dots (5)$$

$$(r_s)_3 = \frac{a}{2} \sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{5}{N}\right)^2} \dots \dots \dots (6)$$

and

$$(r_s)_4 = \frac{a}{2} \sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{7}{N}\right)^2} \dots \dots \dots (7)$$

In like manner, from inspection of Fig. 7, and using the relation:

$$n = \frac{c}{N} \dots \dots \dots (8)$$

We obtain:

$$(r_s)_5 = \frac{a}{2} \sqrt{1 + \left(\frac{7}{NT}\right)^2} \dots \dots \dots (9)$$

$$(r_s)_6 = \frac{a}{2} \sqrt{1 + \left(\frac{5}{NT}\right)^2} \dots \dots \dots (10)$$

$$(r_s)_7 = \frac{a}{2} \sqrt{1 + \left(\frac{3}{NT}\right)^2} \dots \dots \dots (11)$$

and

$$(r_s)_8 = \frac{a}{2} \sqrt{1 + \left(\frac{1}{NT}\right)^2} \dots \dots \dots (12)$$

The radius ratio of the first sector is then

$$R_1 = \frac{(r_s)_1}{r_1} \dots \dots \dots (13)$$

and values of  $R$  for all other sectors are similarly obtained.

The fin plate area ( $S$ ) of each sector is computed as follows:

$$S_1 = [(r_2)_1^2 - r_1^2] \tan^{-1} \left( \frac{2m}{c} \right) \dots \dots \dots (14)$$

Where  $\tan^{-1} \left( \frac{2m}{c} \right)$  is expressed in radians.

Substituting values of  $c$ ,  $m$ , and  $R$ , from Equations (2), (3) and (13), respectively, in Equation (14) and simplifying

$$S_1 = r_1^2 [R_1^2 - 1] \tan^{-1} \left( \frac{2T}{N} \right) \dots \dots \dots (15)$$

Similarly,

$$S_2 = r_1^2 [R_2^2 - 1] \left[ \tan^{-1} \left( \frac{4T}{N} \right) - \tan^{-1} \left( \frac{2T}{N} \right) \right] \dots \dots \dots (16)$$

$$S_3 = r_1^2 [R_3^2 - 1] \left[ \tan^{-1} \left( \frac{6T}{N} \right) - \tan^{-1} \left( \frac{4T}{N} \right) \right] \dots \dots \dots (17)$$

$$S_4 = r_1^2 [R_4^2 - 1] \left[ \tan^{-1} \left( \frac{8T}{N} \right) - \tan^{-1} \left( \frac{6T}{N} \right) \right] \dots \dots \dots (18)$$

By the same procedure, with Equation (8) replacing Equation (3) we obtain:

$$S_5 = r_1^2 [R_5^2 - 1] \left[ \tan^{-1} \left( \frac{8}{NT} \right) - \tan^{-1} \left( \frac{6}{NT} \right) \right] \dots \dots \dots (19)$$

$$S_6 = r_1^2 [R_6^2 - 1] \left[ \tan^{-1} \left( \frac{6}{NT} \right) - \tan^{-1} \left( \frac{4}{NT} \right) \right] \dots \dots \dots (20)$$

$$S_7 = r_1^2 [R_7^2 - 1] \left[ \tan^{-1} \left( \frac{4}{NT} \right) - \tan^{-1} \left( \frac{2}{NT} \right) \right] \dots \dots \dots (21)$$

and

$$S_8 = r_1^2 [R_8^2 - 1] \tan^{-1} \left( \frac{2}{NT} \right) \dots \dots \dots (22)$$

Knowing  $R$  for each sector, and with a given value of  $P$ , the fin effectiveness  $E$  can be determined directly from Fig. 5.

The heat transferred through the first sector is

$$q_1 = h S_1 E_1 \theta_1 \dots \dots \dots (23)$$

so that the total heat transmission through all sectors is

$$q_s = \Sigma q_{1-s} = h \theta_1 [S_1 E_1 + S_2 E_2 + \dots \dots \dots + S_8 E_8] \dots \dots \dots (24)$$

Let

$$q_s = h S_s \bar{E} \theta_1 \dots \dots \dots (25)$$

and

$$S_s = S_1 + S_2 + \dots \dots \dots + S_8 \dots \dots \dots (26)$$

Then the mean effectiveness of the fin surface is represented by:

$$\bar{E} = \frac{S_1 E_1 + S_2 E_2 + \dots \dots \dots + S_8 E_8}{S_s} \dots \dots \dots (27)$$

For the square fin, since  $a = c$

it is only necessary to compute data for the first  $\frac{N}{2}$  sectors, since the square, when

bisected diagonally, consists of two identically disposed fin surface areas.

## APPLICATION

The above methods, for approximation of fin effectiveness ( $E$ ), are illustrated for both a rectangular and square plate fin. The design proportions of fin length and width to tube size represent values that should not be exceeded in good practice. These two examples are selected to demonstrate the maximum error in fin effectiveness ( $E$ ) as encountered in extended surface design.

*Example 1. Rectangular plate fin:*

$$\begin{aligned}a &= 2\frac{1}{2} \text{ in. or } 0.208 \text{ ft} \\c &= 1\frac{1}{2} \text{ in. or } 0.125 \text{ ft} \\r_1 &= \frac{5}{8} \text{ in. or } 0.0260 \text{ ft} \\k &= 220 \text{ Btu/(hr) (sq ft) (deg F/ft)} \\h &= 10 \text{ Btu/(hr) (sq ft) (deg F)} \\b &= 0.010 \text{ in. or } 0.000833 \text{ ft}\end{aligned}$$

$$P = r_1 \sqrt{\frac{2h}{bk}} = 0.026 \sqrt{\frac{2 \times 10}{220 \times 0.000833}} = 0.272$$

$$N = 8 \text{ sectors/quadrant}$$

$$T = \frac{a}{c} = 1.67$$

(a) *Estimation of fin effectiveness by equivalent circular area method:*

$$r_2 = \sqrt{\frac{ac}{\pi}} = \sqrt{\frac{0.208 \times 0.125}{\pi}} = 0.0911 \text{ ft}$$

$$R = \frac{r_2}{r_1} = \frac{0.0911}{0.0260} = 3.51$$

and

$$P = 0.272$$

Hence

$$\bar{E} = 0.772 \text{ (from Fig. 5)}$$

(b) *Estimation of fin effectiveness by segmentation:* Referring to equations of Method No. 2 the data for sector 3 are computed in detail

$$(r_2)_3 = \frac{a}{2} \sqrt{\left(\frac{1}{T}\right)^2 + \left(\frac{5}{N}\right)^2}$$

or

$$(r_2)_3 = \frac{0.208}{2} \sqrt{\left(\frac{1}{1.67}\right)^2 + \left(\frac{5}{8}\right)^2} = 0.0901 \text{ ft}$$

$$r_1 = 0.0260 \text{ ft}$$

$$R_3 = \frac{(r_2)_3}{r_1} = \frac{0.0901}{0.0260} = 3.47$$

$$S_3 = r_1^2 [R_3^2 - 1] \left[ \tan^{-1} \left( \frac{6T}{N} \right) - \tan^{-1} \left( \frac{4T}{N} \right) \right]$$

or

$$S_3 = 0.0260^2 [3.47^2 - 1] \left[ \tan^{-1} \frac{6 \times 1.67}{8} - \tan^{-1} \frac{4 \times 1.67}{8} \right] = 0.00150 \text{ sq ft}$$

Data for other sectors are computed in an identical manner.

The following table lists essential data to determine the mean fin effectiveness,  $\bar{E}$ .

SECTOR NO.	R	P	S	E	SE	$\bar{E}$
1	2.45	0.272	0.00135	0.924	0.00125	—
2	2.84	0.272	0.00143	0.879	0.00126	—
3	3.47	0.272	0.00150	0.788	0.00118	—
4	4.23	0.272	0.00154	0.675	0.00104	—
5	4.52	0.272	0.00155	0.635	0.00098	—
6	4.27	0.272	0.00154	0.670	0.00103	—
7	4.10	0.272	0.00153	0.693	0.00106	—
8	4.02	0.272	0.00153	0.705	0.00108	—
$\Sigma$	...	...	0.01197	...	0.00888	0.742

The mean fin effectiveness, for this fin, as computed by the equivalent circular area method, is high by approximately

$$\frac{(0.772 - 0.742)100}{0.742} = 4 \text{ per cent}$$

This amount of error is probably within limits of accuracy involved in the assumption of a constant fluid convection coefficient ( $h$ ) over the entire fin surface, as used in deriving the expression for fin effectiveness. For practical purposes, the use of the equivalent circular area method gives a sufficiently accurate value of fin effectiveness, although this results in a somewhat higher value than indicated by the more exact method.

*Example 2*—Square plate fin:

$$\begin{aligned} a &= 2 \text{ in. or } 0.167 \text{ ft} \\ c &= 2 \text{ in. or } 0.167 \text{ ft} \\ r_1 &= \frac{1}{8} \text{ in. or } 0.0260 \text{ ft} \\ k &= 220 \text{ Btu/(hr) (sq ft) (deg F/ft)} \\ h &= 10 \text{ Btu/(hr) (sq ft) (deg F)} \\ b &= 0.010 \text{ in. or } 0.000833 \text{ ft} \end{aligned}$$

$$P = r_1 \sqrt{\frac{2h}{bk}} = 0.272$$

$$N = 8 \text{ sectors/quadrant}$$

$$T = \frac{a}{c} = 1$$

(a) *Estimation of fin effectiveness by equivalent circular area method:*

$$r_2 = \sqrt{\frac{ac}{\pi}} = \sqrt{\frac{0.167 \times 0.167}{\pi}} = 0.0940 \text{ ft}$$

$$R = \frac{r_2}{r_1} = \frac{0.094}{0.026} = 3.61$$

and

$$P = 0.272$$

Hence

$$E = 0.768 \text{ (from Fig. 5)}$$

(b) *Estimation of fin effectiveness by segmentation:* The following table lists necessary factors for determining the mean fin effectiveness and are computed as in *Example 1*.

SECTOR NO.	R	P	S	E	SE	$\bar{E}$
1	3.23	0.272	0.00156	0.824	0.00128	—
2	3.43	0.272	0.00160	0.795	0.00127	—
3	3.78	0.272	0.00162	0.743	0.00120	—
4	4.25	0.272	0.00163	0.672	0.00110	—
$\Sigma$	...	...	0.00641	...	0.00485	0.757

The mean fin effectiveness, as computed by the equivalent circular area method, is high by approximately

$$\frac{(0.768 - 0.757) 100}{0.757} = 1.5 \text{ per cent}$$

The method using the equivalent circular area therefore indicates a fin effectiveness of sufficient accuracy to use in practice, when applied to square fin plates.

*Note:* The foregoing method of determining the fin effectiveness by segmentation, based on the assumption of a square or rectangular fin area served by each tube, applies to both *in line* or *staggered* tube arrangements where *individual* fins are assembled on *each* tube. Further, it is applicable to *in line* tube nests, as shown in Fig. 4, where the fin plate is *continuous* over the entire tube nest.

In designs employing a *staggered* tube arrangement with a *continuous* fin plate, the fin area served by a tube is hexagonal. For this type surface arrangement the method of rectangular or square fin segmentation, developed in this Appendix, may be applied in a similar manner to the hexagonal area.

## APPENDIX V

EXAMPLES ILLUSTRATING THE RELATIVE MAGNITUDE OF INDIVIDUAL THERMAL RESISTANCES OF A TYPICAL EXTENDED SURFACE AIR COOLER FOR VARIOUS APPLICATIONS

- $S = S_i + S_o$  = Total outside surface, sq ft/lineal ft tube
- $S_i$  = Inside tube surface, sq ft/lineal ft tube
- $S_o$  = Outside exposed tube surface, sq ft/lineal ft tube
- $S_f$  = Lateral fin surface, sq ft/lineal ft tube
- $a$  = Fin plate length/tube, ft
- $b$  = Fin plate thickness, ft
- $c$  = Fin plate width/tube, ft
- $E$  = Fin effectiveness, dimensionless
- $F_s$  = Ratio of sensible to total heat removed from air, dimensionless
- $h_a$  = Air film coefficient, Btu/(hr) (sq ft of  $S$ ) (deg F)
- $h_i$  = Water film coefficient, Btu/(hr) (sq ft of  $S_i$ ) (deg F)
- $k$  = Fin metal thermal conductivity, Btu/(hr) (sq ft) (deg F/ft)

$$P = r_i \sqrt{\frac{2k}{kb}}, \text{ dimensionless}$$

$$r_i^1 = \text{Inside tube radius, ft}$$

$$r_i = \text{Outside tube radius, ft}$$

$$r_s = \sqrt{\frac{ac}{\pi}} = \text{Radius of equivalent circular fin area, ft}$$

$$R = r_s/r_i, \text{ dimensionless}$$

$$R_{os} = \text{Overall thermal resistance to sensible heat flow, Btu/(hr) (sq ft of } S \text{) (deg F overall)}$$

$$R_a = \frac{1}{h_a} = \text{Air film thermal resistance, Btu/(hr) (sq ft of } S \text{) (deg F)}$$

$$R_m = \frac{1}{h_a} \left( \frac{1 - E}{E + \frac{S_p}{S_t}} \right) = \text{Metal thermal resistance, Btu/(hr) (sq ft of } S) (\text{deg F})$$

$$R_1 = \frac{S}{S_1 h_1} = \text{Water film thermal resistance, Btu/(hr) (sq ft of } S) (\text{deg F})$$

**Surface Used for Illustration:** A typical, extended surface air cooler, used commercially, has been selected for illustrative purposes. The fin surface consists of rectangular, flat plates bonded to round tubes. In the following examples air, flowing over the fin plates, is cooled by water passing through the tubes. The surface is nested with tubes *in line* in adjacent rows.

The surface physical data are as follows:

$$a = 1-11/16 \text{ in. or } 0.141 \text{ ft}$$

$$b = 0.010 \text{ in. or } 0.000833 \text{ ft}$$

$$c = 1-1/2 \text{ in. or } 0.125 \text{ ft}$$

$$r_1 = 1/4 \text{ in. or } 0.0209 \text{ ft}$$

$$r^1 = 0.222 \text{ in. or } 0.0185 \text{ ft}$$

$$r_2 = \sqrt{\frac{ac}{\pi}} = 0.0749 \text{ ft}$$

$$R = r_2/r_1 = 3.60$$

$$S_1 = 0.116 \text{ sq ft/lineal ft tube}$$

$$S_t = 2.33 \text{ sq ft/lineal ft tube}$$

$$S_p = 0.12 \text{ sq ft/lineal ft tube}$$

$$\bar{S} = 2.45 \text{ sq ft/lineal ft tube}$$

$$\text{Fin Spacing} = 6/\text{in.}$$

$$\text{Tube Material} = \text{Copper}$$

$$\text{Fin Material} = \text{Copper}$$

**Thermal Resistance Formula:** In the following formula for the overall sensible heat thermal resistance ( $R_{os}$ ), the thermal resistance through the prime tube wall is neglected, and a perfect thermal bond assumed between the fin base and outside tube wall. The individual thermal resistances, referred to the total outside surface area ( $S$ ), form the overall sensible heat thermal resistance ( $R_{os}$ ) as follows:

$$R_{os} = R_a + \frac{R_m + R_1}{F_s} \dots \dots \dots (1)$$

**Scope of Examples:** Examples 1 and 2 compare the typical heat transfer performance of free versus forced air convection applications, respectively. No air latent heat removal is considered in either of these examples.

**Example 3** indicates the heat transfer performance where some moisture is removed from the chilled air. This latter example represents a normal application for surface dehumidifiers.

**Value of tube and air side transfer coefficients:** The tube side (water) heat transfer coefficient ( $h_1$ ) is selected for approximately 3 ft per second water velocity at normal chilled water temperatures. The two values of the air film transfer coefficient ( $h_a$ ) illustrate the difference between free and forced air convection applications. Table 1 lists the values of both type coefficients used in these comparisons.

TABLE 1—VALUE OF BOTH TYPES OF TRANSFER COEFFICIENTS USED

EXAMPLE NO.	1	2	3
$h_a$ .....	4	9	9
$h_1$ .....	600	600	600

**Evaluation of metal thermal resistance ( $R_m$ ):** The fin effectiveness ( $E$ ) is evaluated by using the equivalent circular area method. (See Appendix IV). The steps in determining both  $E$  and  $R_m$  are summarized in Table 2.

TABLE 2—DETERMINATION OF  $E$  AND  $R_m$ 

EXAMPLE NO.	1	2 AND 3
$r_2$ .....	0.0749 ft	0.0749 ft
$r_1$ .....	0.0208 ft	0.0208 ft
$h_a$ .....	4	9
$h$ .....	220	220
$b$ .....	0.000833 ft	0.000833 ft
$P = r_1 \sqrt{\frac{2h_a}{bk}}$ .....	0.138	0.207
$R = r_2/r_1$ .....	3.60	3.60
$E$ from Fig. 5.....	0.925	0.849
$S_p$ .....	0.12	0.12
$S_t$ .....	2.33	2.33
$R_m$ .....	0.0192	0.0186

Note from Table 2 the small variation of  $R_m$  over a wide range of the air film coefficient ( $h_a$ ). For this specific fin,  $R_m$  changes approximately 3 per cent for a 225 per cent change of  $h_a$ , while  $E$  varies about 9 per cent.

Relative magnitudes of individual thermal resistances for various applications: Table 3 is compiled by use of Equation (1) and data in Tables 1 and 2.

TABLE 3—RELATIVE MAGNITUDES OF INDIVIDUAL THERMAL RESISTANCES FOR VARIOUS APPLICATIONS

EXAMPLE NO.	1	2	3
Application.....	Free Conv. (Dry)	Forced Conv. (Dry)	Forced Conv. (Dehumid- ifying)
$F_a$ .....	1.0	1.0	0.7
$R_a$ .....	0.250	0.111	0.111
$R_m/F_a$ .....	0.0192	0.0186	0.0266
$R_1/F_a$ .....	0.0352	0.0352	0.0503
$R_{os}$ .....	0.3044	0.1648	0.1879
100 ( $R_a/R_{os}$ ) = per cent air film resistance.....	82.1	67.3	59.0
100 ( $R_m/R_{os} \times F_a$ ) = per cent metal resistance.....	6.3	11.3	14.2
100 ( $R_1/R_{os} \times F_a$ ) = per cent water film resistance.....	11.6	21.4	26.8

The last three lines in Table 3 list the values of the individual thermal resistances, expressed as a percentage of the overall sensible heat thermal resistance. The metal or fin plate represents from approximately 6 per cent to 14 per cent of the total resistance to heat flow, for this specific coil design.

Thus a knowledge of the metal resistance is of importance in determining the thermal performance of extended surfaces. The metal resistance may or may not be an appreciable part of the total thermal resistance depending upon combinations of the following design variables for the specific transfer surface:

- |          |          |          |            |
|----------|----------|----------|------------|
| 1. $r_1$ | 3. $h_a$ | 5. $h$   | 7. $S/S_1$ |
| 2. $r_2$ | 4. $b$   | 6. $h_1$ | 8. $F_a$   |

## APPENDIX VI

### COMPARISON OF HEATER PERFORMANCE

It is frequently required to compare the performance of one heater with another of different design or proportions. These heaters are generally tested at different air velocities but generally the temperature rise and the resistance to air flow will not correspond. The values for  $U$  or the overall rate of heat transfer may or may not be the

same. The commercial value of a heater per square foot of surface depends upon the value of  $U$  under some fixed condition at which it is supposed to operate. These fixed conditions are ratio of initial to final temperature difference and pressure drop when handling a fixed quantity of air. The usual method of approach to this problem has been to draw graphs of performance and interpolate for corresponding conditions for two different heaters. This has been well outlined by Stacey and Ashley<sup>6</sup> but it is also possible to do this without graphs where  $h$ , the coefficient of heat transfer, and the pressure drop,  $\Delta p$ , for a given arrangement of heater and air quantity, is known. In comparing the economy of two different designs of plate fins, for example, it is necessary to have a convenient method for making this comparison. Such a method has been worked out and has been in use, in effect, for some time. Since the use of this method is an essential part in the design study of heaters, its presentation here is perhaps in order. The following symbols are employed:

- $S$  = square foot of heater surface.  
 $A$  = face area of heater—square feet.  
 $L$  = relative depth of heater.  
 $G$  = pounds of air per hour passing through the heater.  
 $V$  = air velocity through face area of heater  
 $C_p$  = specific heat of the air or change in total heat content per degree change in temperature.  
 $\rho$  = air density—pounds per cubic foot.  
 $\Delta p$  = drop in air pressure passing through the heater at the velocity  $V$ —inches water gage.  
 $t_s$  = temperature of heating or cooling fluid  
 $t_1$  = temperature of entering air.  
 $t_2$  = temperature of leaving air.  
 $\theta_1 = t_s - t_1$ .  
 $\theta_2 = t_s - t_2$ .  
 $R = \left( \frac{\theta_1}{\theta_2} \right)$   
 $U$  = overall coefficient of heat transfer Btu per hour per square foot per degree difference.

Perhaps the simplest method of approach to this problem is as follows: Assume a heat transfer surface of flexible arrangement as to depth ( $L$ ) and area ( $A$ ) but constant surface ( $S$ ) so that  $AL$  is constant for all arrangements. We may now show that by change in  $A$ ,  $L$  and  $G$  we may simulate any performance in respect to  $R$  and  $\Delta p$ .

There are three primary equations which relate the independent variables  $\Delta p$ ,  $R$  and  $V$ . These are:

$$\log R = \frac{US}{C_p G} \quad (1)$$

$$\Delta p = C_1 V^n L \quad (2)$$

where

$C_1$  and  $n$  = constants for any given type of heater surface,  $n$  = a value approaching 2 but always slightly less than 2.

$$U = C_2 V^m \quad (3)$$

where

$C_2$  and  $m$  = empirical values determined by test for a given type of heater, and may both be taken as constant through a velocity range not exceeding a ratio of 2 to 1.

Also, we have the following dimensional relationships based on the assumption that the surface  $S$  remains constant but that the face area  $A$  and the depth of the heater  $L$  are varied to simulate any desired performance in respect to  $R$  and  $\Delta p$ .

$$AL = C_3, \text{ a constant since } S \text{ is constant for this assumption} \quad (4)$$

$$V = \frac{G}{60\rho A} \text{ or } G = 60AV\rho \quad (5)$$

<sup>6</sup>Loc. cit. See Note 2.

Since we are to compare two arrangements which we may designate as (1) and (2), we will employ the ratios of the various values rather than the absolute values. For convenience, we may express the ratios

$$\frac{\log R_2}{\log R_1} \frac{\Delta p_2}{\Delta p_1} \frac{A_2}{A_1}, \text{ etc., simply as}$$

$\log R_2, \Delta p_2, A_2$ , etc. Then we have from Equations (1), (2) and (3)

$$\log R_2 = \frac{U_2}{G_2} \dots \dots \dots (6)$$

$$\Delta p_2 = V_2^n L_2 \text{ and} \dots \dots \dots (7)$$

$$U_2 = V_2^m \dots \dots \dots (8)$$

From Equations (4) and (5) we have

$$A_2 L_2 = 1 \dots \dots \dots (9)$$

and

$$G_2 = A_2 V_2 \dots \dots \dots (10)$$

Combining Equation (6) with Equation (10), we have

$$\log R_2 = \frac{V_2^m L_2}{V_2} = \frac{L_2}{V_2^{(1-m)}} \dots \dots \dots (11)$$

or

$$L_2 = V_2^{(1-m)} \log R_2 \dots \dots \dots (12)$$

From Equation (7), we have

$$L_2 = \frac{\Delta p_2}{V_2^n} \dots \dots \dots (13)$$

Equating Equations (12) and (13), we have

$$V_2^{(1-m)} \log R_2 = \frac{\Delta p_2}{V_2^n} \dots \dots \dots (14)$$

$$V_2^{(1+n-m)} = \frac{\Delta p_2}{\log R_2} \dots \dots \dots (15)$$

Combining Equation (15) with Equation (8), we have

$$U_2 = \left( \frac{\Delta p_2}{\log R_2} \right)^{m/(1+n-m)} \dots \dots \dots (16)$$

That is

$$\frac{U_2}{U_1} = \left[ \frac{\Delta p_2 / \Delta p_1}{\log R_2 / \log R_1} \right]^{m/(1+n-m)} \text{ Also we find}$$

$$\frac{(S/A)_2}{(S/A)_1} = \left( \frac{U_1}{U_2} \right) \left( \frac{V_2}{V_1} \right) \dots \dots \dots (17)$$

Equation (16) now gives us the comparative value of  $U_2$  to  $U_1$  for the desired values of  $\Delta p_2$  and  $R_2$ . We can now compare the heater B directly with the heater A since they both have the same pressure drop and the same ratio of initial to final temperature difference. That is, they are compared under conditions of equal performance. If  $U_a$  is the coefficient of heat transfer of heater A and  $U_{b2}$  is the heat transfer coefficient for heater B under conditions of equal performance as in Equation (16), then the ratio  $\frac{U_{b2}}{U_a}$  is the relative value of the two surfaces per square foot.

Generally, the exponents  $m$  and  $n$  can each be determined from two test points on heater B if these test points are determined with sufficient accuracy or lie on a test

curve. The two test points chosen should, in each case, bracket the temperatures and pressures which are to be compared. If two test points of pressure  $\Delta p_1$  and  $\Delta p_2$  are determined for corresponding velocities  $V_1$  and  $V_2$ , then the exponent  $n =$

$$\log \left( \frac{\Delta p_2}{\Delta p_1} \right) / \log \left( \frac{V_2}{V_1} \right) \text{ and the exponent for heat transfer } m = \log \left( \frac{U_2}{U_1} \right) / \log \left( \frac{V_2}{V_1} \right).$$

Thus while  $m$  and  $n$  may vary at different points of the curve, yet the values of  $m$  and  $n$  that apply to the correction in Equation (16) are obtained with sufficient accuracy so that the desired value  $U_2$  or  $\frac{U_2}{U_1}$  is as accurate as any test value would be for the heater

in which the dimensions were changed. It is understood, of course, that  $\Delta p_2$  and  $U_2$  are determined directly from tests on heater B while  $\Delta p_1$  and  $U_1$  are hypothetical calculated values which by rearrangement of heater B would give the same results as that obtained by heater A. An example will make this clear.

*Example:* Assume we have a heater which we will designate as design A for which we have tabulated performance values against which we wish to compare the corresponding performance of a different design, B; and that we have at least two accurate test points of performance for the latter. What is the relative utility value of a square foot of surface of B as compared with a square foot of surface of A? Assume we select one point of performance of heater A which is bracketed by the two test points for heater B and that we have the following performance data:

For A—

$$V_A = 500 \text{ ft/min} = \text{velocity of standard air over face area.}$$

$$(\Delta p)_A = 0.4 \text{ in.}$$

$$\theta_{1A} = t_s - t_1 = 100 = \text{difference between the temperature of the heating fluid and the entering air.}$$

$$\theta_{2A} = t_s - t_2 = 30 = \text{difference in temperature between the heating fluid and the leaving air.}$$

$$R_A = \left( \frac{\theta_{1A}}{\theta_{2A}} \right) = 3.33$$

$$C_p = 0.24$$

$$U_A = 10$$

$$(S/A)_A = 65.0$$

*First Test Point (for B)*

$$V_1 = 540 \text{ ft/min}$$

$$\Delta p_1 = 0.5 \text{ in. (water gage)}$$

$$\theta_1 = 100 \text{ degrees}$$

$$\theta_2 = 40 \text{ degrees}$$

$$R_1 = \left( \frac{\theta_1}{\theta_2} \right) = 2.5$$

$$C_p = 0.24$$

$$(S/A)_B = 56.4 \text{ sq ft/sq ft}$$

$$U_1 = \frac{60 C_p V_1 \log R_1}{(S/A)_B} = \frac{1.08 \times 540 \times 0.915}{56.4} = 9.47$$

*Second Test Point (for B)*

$$V_2 = 360$$

$$\Delta p_2 = 0.24 \text{ in.}$$

$$\theta_1 = 90 \text{ degrees}$$

$$\theta_2 = 30 \text{ degrees}$$

$$R_2 = \left( \frac{\theta_1}{\theta_2} \right) = 3.0$$

$$C_p = 0.24$$

$$(S/A)_B = 56.4$$

$$U_2 = \frac{1.08 \times 360 \times 1.099}{56.4} = 7.58$$

*From B—First and Second Tests*

$$\left( \frac{V_2}{V_1} \right)^m = \frac{U_2}{U_1} = \left( \frac{360}{540} \right)^m = \left( \frac{7.58}{9.47} \right) = 0.80$$

$$m = .55$$

$$\left(\frac{V_1}{V_2}\right)^n = \frac{\Delta p_2}{\Delta p_1} = \left(\frac{360}{540}\right)^n = \left(\frac{0.24}{0.50}\right)$$

$$n = 1.805$$

Having thus established  $m = 0.55$  and  $n = 1.805$  from the two tests for design B, and taking from the first test point  $R_B = \left(\frac{\theta_1}{\theta_2}\right) = 2.5$ ,  $\Delta p_B = 0.5$  in.,  $U_B = 9.47$ ,  $V_B = 540$ , we may calculate  $U_{BC}$  and  $V_{BC}$  for a new arrangement  $(S/A)_{BC}$  which will give  $R_{BC} = R_A = 3.33$  and  $(\Delta p)_{BC} = (\Delta p)_A = 0.4$  in.

The equation giving  $U_{BC}$  (see Equation 16) is

$$\frac{U_{BC}}{U_{B1}} = \left[ \frac{(\Delta p)_{BC}/(\Delta p)_{B1}}{\log R_{BC}/\log R_{B1}} \right]^{m/(1+n-m)} = \left( \frac{0.80}{1.33} \right)^{0.344} = 0.883$$

$$U_{BC} = 9.47 \times 0.883 = 8.36$$

We are now able to compare the value of the B surface with that of the A surface. This ratio of value under conditions of equal performance is

$$100 \left( \frac{8.36}{10} \right) = 83.6 \text{ per cent} = \frac{\text{Value of sq ft B surface}}{\text{Value of sq ft A surface}}$$

$$V_{BC} = 540 \times \left( \frac{8.36}{9.47} \right)^{(1/0.55)} = 430 \text{ ft/min}$$

Corresponding to Equation (17), the formula giving  $(S/A)_{BC}$  is

$$(S/A)_{BC} = (S/A)_A \times \left( \frac{V_{BC}}{V_A} \right) \times \left( \frac{U_A}{U_{BC}} \right)$$

$$(S/A)_{BC} = 65.0 \times \left( \frac{430}{500} \right) \times \left( \frac{10}{8.36} \right) = 66.9 \text{ sq ft/sq ft}$$

## DISCUSSION

H. B. NOTTAGE, East Hartford, Conn.: Some time ago, when Dr. Carrier was engaged in completing the first formulations of the problem discussed in this paper, it was my fortunate privilege to be given the opportunity of reviewing the analytical details involved. The series solution to the basic differential equation most definitely seemed to offer practical possibilities of effective application.

My particular interest dealt with a general approach to the case of a circumferential fin with tapered sides, on a base cylinder of fairly large diameter. In further regard, I was concerned with the design problem for a family of rather closely-spaced fins, rather than an individual isolated fin. It is to be recognized that there is no exact analytical solution available for the basic differential equation describing this case. A series solution offers one type of approximation method; but the use of a series implies an obligation to also bring forth a concise coverage of the limitations on range and accuracy inherent in all such approximations. No doubt, the author's solution is adequate for most engineering purposes, and what are its limitations?

Following my initial discussion with Dr. Carrier, I was fortunate enough to procure the services of a physicist and a mathematician in seeking a precise evaluation of the merits of the series solution. The net result of these efforts was a judgment that the series method was adequate only for fins of fairly high effectiveness and slight taper. For the general case, there was recommended to me, instead, a solution of the basic differential equation based upon Picard's method of successive approximations. When applied to the problem of my particular interest, this latter method was demonstrated

to have a more rapid convergence and improved accuracy over the series solution; although this is no serious point against the series method when properly delimited.

Further comments on the assumptions invoked by the authors are in order.

It was stated that the solution obtained required the fin base temperature to be known, and it was further implied that this temperature should be the same at all points. I do not necessarily feel that such a limitation is essential. It is possible to extend the solution of the problem by eliminating the base temperature and its variations from point to point, in terms of the independent variables which establish this temperature. This extension involves going into the fluid conditions on the prime-surface side of the exchanger, and includes a consideration of the heat conduction through the base metal. The heat transfer from the exposed surface of the base metal is also properly included through this extension of the general problem; and this term is not negligible in many instances.

Another serious limitation on the extensive application of this analysis, which is common to most of those which are set up to deal with fin problems, lies in the assumption that the unit surface conductance for heat transfer is constant at some mean effective value over the entire surface. This is done for want of adequate data on point-to-point conditions for the heat transfer and flow processes. The effect is usually blanketed out in the reduction of over-all experimental data, followed by the re-application of such data to design problems on an over-all basis. However, it has been my experience that this unit surface conductance *cannot* be treated as a mean constant quantity over a wide range of geometrical and heat transfer conditions. There is a very real point-to-point variation here, which must be recognized and accounted for, to permit design to close limits over all extremes of operating conditions. I cannot emphasize too strongly the present need for patient and thorough experimental data covering problems of this nature.

The idealization that all of the heat flow through the fins is radially outward requires the isotherms in the metal to be concentric circles. This would not be expected always; although apparently the analysis is not too sensitive to departures therefrom. The mathematics of combined radial and circumferential heat flow are difficult for all but the simpler cases.

Then, there arises the question of performance comparisons. The authors have dealt with over-all comparisons only. These regard the heat exchanger as a *box*, through which streams flow, and for which certain hypotheses are made regarding the internal heat exchange process. Then, convenient empirical exponents are applied, based upon data which are usually rather limited and not necessarily too accurate. These methods are workable for the usual interpolation comparisons and quick estimates; but for design problems where extrapolation is required, a breakdown into the component effects influencing the over-all performance is desirable. It is my impression that often empirical exponents are trusted too far under inappropriate circumstances.

When we come to comparisons dealing with the merits of the component process of an over-all heat exchange, a first interest is in an evaluation of the performance of the fluid-swept surface itself, in its form, finish, and extent. For instance, the crimped fin mentioned by the authors has an irregular or rough surface. How is a true judgment of this construction to be obtained upon an internal rather than an over-all basis? A method of doing this has recently been proposed by Dr. A. P. Colburn.<sup>7</sup> In effect, his suggested comparison interrelates the unit surface conductance for heat transfer and the corresponding power loss in the fluid flow stream per unit of surface area swept. This, I have found very useful, myself.

By way of conclusion, I would like to compliment the authors for their contributions to an important problem. The fin effectiveness correlation presented in this paper

<sup>7</sup> Heat Transfer by Natural and Forced Convection, by A. P. Colburn. (*Purdue University Engineering Bulletin*, Vol. XXVI, No. 1, January, 1942.)

should aid greatly the understanding of such extended-surface heat exchanger units as are employed in heating, ventilating, and air conditioning equipment.

G. L. TUVE, Cleveland, Ohio: This sounds a bit complicated, but actually it is a simplification of the problem compared with what we have had before, and I think the authors are to be complimented. I hope it will work out so that the three types of resistance that we recognize in a fin coil can all be taken care of easily. We have an air side resistance, a fin effectiveness, and an inside surface coefficient resistance. The air side resistance problem has been pretty well aired and after the war, when it can be discussed, I think that will be cleared up.

Nobody knew just what to say about fin effectiveness. The authors are to be complimented on simplifying this phase of the problem. The inside surface coefficient is complicated for one case; namely, that of evaporating refrigerants. The Society is now initiating some research along that line that I hope may help to clear it up.

We are very much in need of a standard test code for blast coils, preferably one that is simpler than the code proposed by the *Blast Coil Manufacturer's Institute*. General agreement upon a simplified method for computing blast coil performance would be highly desirable, and this paper is an important contribution toward that goal.

WILLIAM GOODMAN, LaCrosse, Wis.: To many of us the principal value of *fin effectiveness* is that it enables us to predict the performance of various coils in advance. To do so we must use the air film coefficient for prime surface and the fin effectiveness together. Then with the inside surface film coefficient known, the over-all film coefficient can be predicted. Similarly, to determine the true air film coefficient from a test on a finned coil, accurate values of the *fin effectiveness* are also required.

However, in order either to predict the over-all coefficient of a finned coil or to determine the actual value of the film coefficient from a test on a finned coil, we must be certain that the external film coefficient—rather, the air film coefficient, is not affected by the presence of the fins. In some tests reported in a European paper a few years ago it was found that fin spacing did have a marked effect on the film coefficient. As long as the fins were widely spaced along the tube the film coefficient was not affected by the presence of the fins, and therefore, the effectiveness could be used to predict quite accurately the over-all film coefficient; but when the fins were crowded together and the spacing between the fins was reduced to less than approximately one-third of the tube diameter, then the results began to disagree. The over-all coefficient determined by experiments was usually lower than the result obtained by computation.

In all our work, at least in commercial coils, the fins are much closer than one-third of the tube diameter. They are frequently considerably less than that, so that in applying the *fin effectiveness* we have to be careful.

I do not mean in any way to detract from the authors' work, because the accurate determination of fin effectiveness is an important step, and one which they seem to have accomplished, but I do think that further work is necessary in applying these fin effectiveness values to closely spaced fins.

DR. CARRIER: With reference to Mr. Nottage's comments, I was very much interested in the fact that he had referred this to mathematicians and physicists because it is really quite a difficult mathematical and physical problem. I note that he says they questioned the accuracy of the series method of solution. Qualitatively the mathematician would be quite correct as, for example, in the series of values shown by the curves. If we were to use values beyond that shown in the curves or even the more extreme values given in the curves, the error might be appreciable if an insufficient number of terms were employed without a correction for the remaining terms. Fortunately, however, the values in which we are interested from a practical standpoint never lie in this extreme region. Furthermore, we did take particular

precautions to investigate the probable residual value of the remaining terms. In every instance, we calculated eight terms and in a few cases to determine the trend of the residual we calculated nine or ten terms. A rather ingenious method was also worked out which seemed to predict quite accurately the summation of the remaining terms in the infinite series. This method appeared to be quite accurate so long as the series remained reasonably convergent. This method depended upon a study of the decrements of the unknown residual with successive terms up to ten. The law of decrease was found to be a straight line logarithmic function within the range that we could test it. Therefore, it became possible to integrate the residual mathematically on the assumption that this logarithmic decrement continued. This, of course, assumed an extrapolation which could not be proven beyond the ten terms examined. However, in the calculation of all practical physical arrangements the value of this decrement was so small as to not be of serious consequence if it were omitted and the error in its calculation could in no way be large. The curves which are published in this paper contain the correction for the residual and there is no possibility by any other method of calculation of finding results which will differ from those shown by more than an insignificant percentage. In other words, within the limits shown the results are correct by any method of integration.

There was another point of considerable interest that was made by Mr. Nottage. This was the effect that the varying values of  $h$  (surface coefficient of heat transfer) might have on the apparent metal resistance since the value of  $h$  appears in equation of the metal resistance. It is true, of course, that a change in the value of  $h$  at different parts of the surface would distort the lines of heat flow as would also the change in temperature of the air passing over the surface. The distorted lines of heat flow would be longer than the straight lines and not radial as assumed. The answer to this is that in calculating the resistance from the effectiveness curves, the value of  $h$  appears both in the numerator and the denominator of a fraction so that in calculating the resistance of the metal on the basis of the resistance formula, variations in  $h$  largely iron out or cancel, although not entirely. It is for this reason that approximate test values of  $h$  or even approximately assumed values of  $h$  do not greatly change the calculated value of resistance of the fins although they do change considerably the value of a calculated effective temperature difference.

It would perhaps have been better to have drawn a plot in terms of resistance rather than in terms of effective temperature difference since the former would show but little change with the value of  $h$ . The average values of  $h$  are, of course, determined as Mr. Goodman indicates from actual tests in which other variables may be evaluated and then eliminated. This procedure gives us practical average values of  $h$  which do not vary widely for small changes in design. The greatest change would probably be caused by the change in the spacing of the fins and for any large change in spacing the value of  $h$  would have to be determined by test unless the flow in all cases was in the nonturbulent region. In the latter case,  $h$  would vary approximately inversely as the spacing. In the turbulent region, the value of  $h$  should follow closely the square root of the coefficient of friction which can be calculated with reasonable accuracy from known data.

As for the method of comparison of the commercial value of two different types of heat transfer surface, there are as indicated in the paper two methods which may be employed. The older method, which is also quite correct, is by interpolation of plots in a family of curves for both types of heaters. One of these plots is superimposed on the other and the point of common performance is found on each. This method is correct but the method which I have proposed has a considerable advantage in comparing a single point of performance for the two units and is equally accurate if we assume no observational errors. It is well known, however, that different heaters will compare somewhat differently in different ranges of performance and if the entire range of performance is to be examined, then the method of calculation has but little if any advantage over the graphical method.

It should be observed that the over-all value of heat transfer  $U$  is the only value that has any significance in comparing performance of two different types of units. In analysis for design, the heat transfer coefficients or resistance must be separated as component parts of the over-all value  $U$  but it should be noted that a heater may have a very low metal resistance or a very high value of  $h$  and still be a poor heater because of the remaining resistance being excessive. It is the comparison of the final designs in relation to their practical performance that alone has commercial importance.



**1248**

# CONTROL OF AIR-STREAMS IN LARGE SPACES

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at Case School of Applied Science.

## INTRODUCTION AND OBJECT

WHEN one or more streams of air are projected into a room or space, for ventilation, heating, cooling or any other purpose, the designer of the system needs basic information on the behavior and control of these streams. A program for the securing of this basic information is being directed by the A.S.H.V.E. Technical Advisory Committee on Air Distribution and Air Friction. This paper is the second report dealing with the characteristics of a primary air-stream in an unconfined space, *i.e.*, a space large enough so that the primary stream is not disturbed by contact with surfaces, or by adjacent streams.<sup>1</sup> The designer has a wide choice of air distribution patterns by varying the size, shape and type of approach to the air discharge outlet, and by controlling the velocity and temperature of the air supply. Present knowledge dealing with the performance of such air-streams is scattered and incomplete. This paper presents experimental evidence in support of certain methods for calculating this air-stream performance.

## SCOPE AND METHODS

Considering the behavior of a free air stream as a jet pump, it was demonstrated in a previous paper<sup>2</sup> that the theory of conservation of momentum could be satisfactorily applied. Experimental results were presented, showing that for every foot of travel of the primary stream away from the outlet, a fixed percentage of room air is entrained, and this percentage was found to be independent of the outlet velocity. These results were based on about 20 common sizes and shapes of outlets, with face velocities of 400 to 2400 fpm, and at distances from the outlet up to about 30 diameters.

Additional studies have now been made with outlet velocities up to 7000 fpm, aspect ratios from 1 to 128, outlet areas from 4 sq in. to 100 sq in. and both rounded-entrance and square-edged approach to the outlet. Long slots have been studied, as well as rectangular, square and round outlets. Readings of air-stream velocity have been taken from 3 ft to 90 ft distant from the discharge face.

One of the disadvantages of the momentum theory and similar methods for calculating air stream performance is that these calculations deal with

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<sup>1</sup> A.S.H.V.E. Research Report No. 1204—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942.)

<sup>2</sup> Loc. Cit. Note 1.

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average velocities, whereas in practice the *maximum* velocity within the stream is likely to be more important. A study has therefore been made to determine what maximum velocity is continuously maintained within any cross section of the stream. For given outlet conditions it is desired to know at what cross section of the stream in the room the velocity does not exceed say 100 fpm, or 500 fpm. The range from 100 fpm to 1000 fpm was extensively studied for all shapes, sizes and outlet velocities included in Table 1.

Some results have been obtained with single long slots, two slots in parallel (one above the other), and multiple slots side by side; also with grilles of

TABLE 1—LIST OF TESTS FOR MAXIMUM RESIDUAL VELOCITY IN AIR STREAMS  
(High Velocity Series)

TEST NO.	NO. OF READINGS	SIZE OF OUTLET INCHES	TYPE OF OUTLET	AREA SQ Ft	RANGE OF OUTLET VELOCITIES	
					Min. fpm	Max. fpm
4 & 5	113	10 x 10	Square Nozzle	0.694	1092	6750
7	112	8½ x 8½	Square Nozzle	0.50	1295	6430
9	82	6 x 6	Square Nozzle	0.25	1390	6740
11	90	4¼ x 4¼	Square Nozzle	0.125	1408	6720
12	72	3 x 3	Square Nozzle	0.062	2150	7010
15	62	2½ x 2½	Square Nozzle	0.031	2820	6700
6	87	25 x 4	Slot Nozzle	0.694	1264	6260
8	87	24 x 3	Slot Nozzle	0.50	1580	6900
10	90	24 x 1½	Slot Nozzle	0.251	1395	6900
13	100	24 x ¾	Slot Nozzle	0.125	1275	6850
14	81	24 x ¾	Slot Nozzle	0.062	2400	6930
16	52	24 x ¾	Slot Nozzle	0.031	1600	6850
17	80	11¾	Round Nozzle	0.769	1405	6600
18	88	8¾	Round Nozzle	0.371	1318	7060
19	90	6	Round Nozzle	0.196	1283	6910
20	84	11¾	Round Orifice	0.769	1282	6760
22	88	8¾	Round Orifice	0.417	1580	7080
21	57	2½	Round Orifice	0.034	1515	7030
22	43	16 x ¾	Slot Nozzle	0.042	2190	7000
23	52	Two 8 x ¾	Slot Nozzle	0.042	1510	7040
24	62	Three 5.3 x ¾	Slot Nozzle	0.042	1725	6990
25	52	Four 4 x ¾	Slot Nozzle	0.042	1820	6920
26	51	Five 3.2 x ¾	Slot Nozzle	0.042	1940	6460

the square-perforated and steel bar types. Work is in progress on long, continuous slots as used in ceiling outlets, and on plates having small perforations. Additional work will be done on heated and cooled air streams projected horizontally or at a slight angle, and on heated and cooled streams diffused by wide-angle grilles.

The entire approach to the problem has been primarily experimental, in an effort to correlate with the theoretical analyses already available in the literature (see section on *Analysis and Correlation*), and to determine the empirical constants or factors necessary for application of theoretical equations.

#### EXPERIMENTAL PROCEDURE

The chief requirements for satisfactory experimental measurements are a well controlled stream from the discharge outlet, a large free open space in

front of the outlet, and suitable methods for measuring air velocities. Most of these tests were made in a large laboratory room about 75 ft wide and 125 ft long with a 40 ft ceiling. The air discharge unit shown in Fig. 1, was located at one end of this laboratory. A similar setup in a smaller room was used for some of the low velocity tests, and for those tests in which the entire cross section of the stream was traversed with a velocity meter as described in the previous paper, see Fig. 2.

Each test was conducted by two observers and each observer independently took readings of the room air velocity with a separate instrument. The general procedure was to establish the required outlet velocity and then make a complete series of observations in the discharge stream, with periodic checks of velocity at the outlet face. In the tests to determine maximum velocities

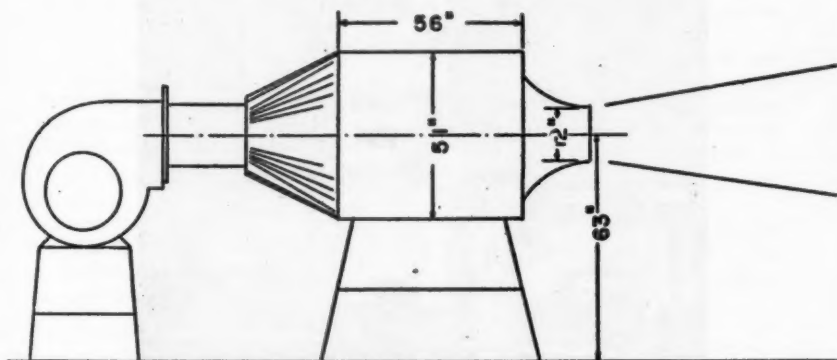


FIG. 1. DIAGRAM OF UNIT USED FOR TESTING OUTLETS LISTED IN TABLE 1.

at various distances from the outlet, the measurements were made at room air velocities of 1000, 500, 400, 250 and 100 fpm. A typical data sheet for one of these tests is shown in Fig. 3.

Preliminary studies were made with various instruments for measuring the room air velocity,<sup>3</sup> but the velometer was found to be the most convenient instrument and hence it was used for most of the tests. Similar tests were also made with an anemometer of the rotating vane-wheel type, but this instrument was not sufficiently sensitive at the lower velocities, although a large number of tests at higher velocities gave good agreement between the velometer and the anemometer. The air-stream velocity by velometer was in all cases the estimated average velometer reading over a period of about one minute, corrected for velometer calibration.

It will be recognized from Fig. 1 that beyond a distance of about 30 ft from the outlet face, there was some interference due to the primary stream striking the floor. This produced higher velocities at or near the floor, but this disturbance did not seem to extend into the stream above the 18 in. level.

<sup>3</sup>A.S.H.V.E. Research Report No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940.)

When measuring the maximum velocities, occurring 5 or 6 ft above the floor, the maximum-velocity curve for a small outlet with a total throw of about 25 ft was identical with the pattern obtained with a large outlet having a total throw of 100 ft. Hence it was decided that the floor interference was not serious as long as the velocities were measured only at the higher level.

In order to determine the relations between maximum velocity and average velocity at any cross section of the stream, a series of tests were made in

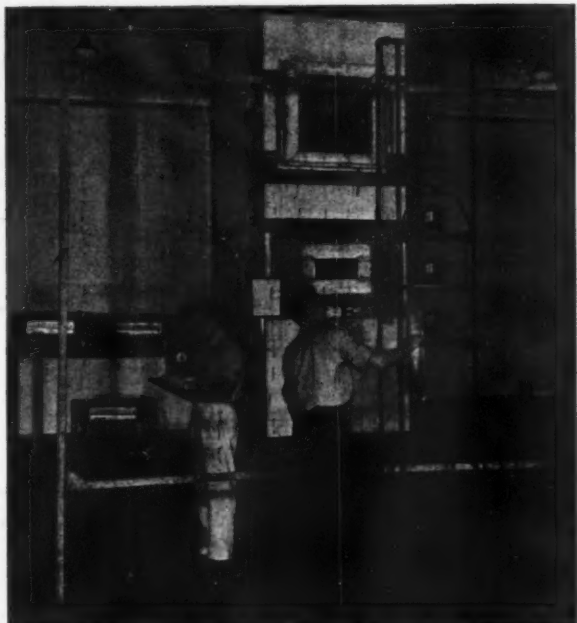


FIG. 2. OBSERVERS CONDUCTING STREAM-TRAVERSE TEST.

the smaller room in which readings were taken at the center of each 6 in. square across the entire air stream, including all velocities above 10 fpm. A large grid of 6 in. squares was formed by threads stretched across a frame, and results were recorded on a data sheet that was ruled as a small scale reproduction of this traverse grid as shown in Fig. 4.

When rounded entrance nozzle outlets were being tested, the outlet velocity was determined by means of an impact tube connected to a sensitive inclined manometer. For all other types of outlets the face velocity was obtained by accurately metering the primary air quantity, and dividing this metered air quantity in cfm by the area of the outlet in square feet. By measuring the static pressure on the inlet side, the *coefficient of discharge* of the various outlets and the true outlet velocity could be computed.

The rounded-entrance outlets of square and rectangular shape were made from heavy wood blocks, and the orifices and round nozzles were of the

standard metering types. The grilles were commercial units mounted in the usual frames. A group of the outlets tested are shown in the photograph, Fig. 5.

### EXPERIMENTAL RESULTS

#### *Maximum Velocity at any Cross Section*

Actual experimental points were plotted on two sets of curves for all the tests listed in Table 1. In both cases the throw or distance from the outlet face was used as the abscissa. In one set of curves, Fig. 6, the residual centerline velocities in the room air stream are plotted on the ordinates, and each curve represents a given outlet velocity. The maximum velocity in the stream decreases as the distance from the outlet face increases. In the other set of curves, Figs. 7 and 8, the various outlet velocities appear on the ordinates, with a curve for each room velocity.

It was recognized that in order to correlate the test results obtained with various sizes and shapes of outlets and types of approach, dimensionless values should be plotted. Hence the ordinates were changed to represent residual velocity in the room in per cent of the outlet velocity, and the throw or  $X$ -distance was measured in *outlet diameters*. Samples of these curves of  $V_r/V_o$  versus  $X/\sqrt{A}$  are shown in Fig. 9. Composite dimensionless curves, on logarithmic coordinates are presented in Fig. 10.

From the results of this series of tests, as shown by the graphs similar to Figs. 6, 7, 8 and 9 (which were plotted for all tests listed in Table 1), it is concluded that the maximum air velocity at any cross section of the stream beyond 25 outlet diameters downstream varies *approximately* as follows:

1. Residual velocity is directly proportional to outlet velocity.
2. Residual velocity is directly proportional to the *effective diameter* of the outlet.
3. Residual velocity is inversely proportional to the distance from the outlet.

If these three statements were exact, the relation could be stated in a very simple equation. Since an examination of Figs. 6 to 10 will show that *these statements are only approximate*, there are two alternatives, (1) Use a more exact equation, and (2) Use the simple equation with a table showing variations in the constant of proportionality. Since the second method is usually preferred by mechanical engineers, it has been adopted here.

#### *Equation for Performance of a Straight-Flow Outlet:*

$$V_r = \frac{CV_o D_o}{X} = \frac{KV_o \sqrt{A_o}}{X} \dots \dots \dots (1a)$$

$$= K \frac{Q}{X\sqrt{A_o}} \dots \dots \dots (1b)$$

where:

- $Q$  = volume of air discharged from outlet, cubic feet per minute.
- $V_r$  = residual maximum velocity in air stream, *i.e.*, the highest maintained velocity at the given cross section in the room.
- $V_o$  = average velocity across the effective area of the outlet.
- = average velocity across measured gross area divided by (coefficient of discharge)  $\times$  (free area).

TABLE 2—VALUES OF  $K$  FOR EQUATION 1

MAXIMUM RESIDUAL VELOCITY	MAXIMUM OUTLET VELOCITIES—FPM				
	1000	2000	3000	4000	5000
500 fpm or above.....	...	6.0	6.2	6.4	6.8
400 fpm.....	...	5.6	5.9	6.2	6.5
300 fpm.....	5.0	5.2	5.4	5.7	6.0
200 fpm.....	4.6	4.8	5.0	5.2	5.4
100 fpm.....	3.7	3.7	3.8	3.9	4.0

TABLE 3—COEFFICIENTS OF DISCHARGE AND VELOCITY RATIOS  
(Free Openings Plenum Approach)

SIZE IN INCHES	NO. OF TESTS	VELOCITY = CFM/AREA	DISTANCE IN FEET	AVG. RATIO OF MAX. VELOCITY TO AVG. VELOCITY	COEFFICIENT OF DISCHARGE
24 x 3	7	400-1200	9	2.6	0.64
18 x 4	3	400-1200	9	2.8	0.58
14.4 x 5	3	400-1200	9	2.4	0.64
12 x 6	6	400-2400	6 & 9	2.7	0.61
10.3 x 7	3	400-1200	9	2.8	0.61
8.5 x 8.5	3	400-1200	9	2.7	0.61
24 x 2	3	600-1200	9	3.2	0.61
24 x 1	8	800-2400	6 & 9	2.8	0.66
24 x 0.5	9	800-2400	6 & 9	2.6	0.67
4.9 x 4.9	8	800-2400	6 & 9	2.9	0.61

TABLE 4—OUTLETS TESTED BY GRID TRAVERSE METHOD

SIZE IN INCHES	TYPE	APPROACH
12 x 6	Free Opening	Duct
12 x 6	Free Opening	Plenum
12 x 6	Bar Grille	Duct
12 x 6	Nozzle Grille	Duct
12 x 6	Square Punch	Duct
20 x 8	Free Opening	Duct
20 x 8	Bar Grille	Duct
21 x 6	Bar Grille	Duct
14 x 4	Free Opening	Duct
12 x 12	Free Opening	Duct
8.5 x 8.5	Free Opening	Plenum
10.29 x 7	Free Opening	Plenum
14.4 x 5	Free Opening	Plenum
18 x 4	Free Opening	Plenum
24 x 3	Free Opening	Plenum
24 x 2	Free Opening	Plenum
24 x 1	Free Opening	Plenum
4.9 x 4.9	Free Opening	Plenum
24 x $\frac{1}{2}$	Free Opening	Plenum
24 x $\frac{1}{4}$	Free Opening	Plenum
24 x $\frac{1}{4}$	Double Slots	Plenum

$A_o$  = effective area of outlet, see Table 5.

= (gross measured area)  $\times$  (free area, decimal)  $\times$  (coefficient of discharge).

$X$  = normal distance from outlet face.

$C$  and  $K$  = constants of proportionality. (Values of  $K$  are given in Table 2.)

### Ratio of Maximum Velocity to Average Velocity

All results given in Figs. 6 to 10 and summarized by Equation (1) apply to the maximum velocity at cross section in the room, parallel to the outlet face.

#### A.S.H.V.E. RESEARCH - AIR-OUTLET THROW DATA

Test No. 18 Outlet Size 8 25" Nominal Outlet Area 53.46  
 Barometer 29.48 Dust Temperature 89° Date 6-26-43  
 Observers KEW-SMF

Pressures		Face Velocity Fpm.	Velometer No. <u>14</u> Actual Reading					Velometer No. <u>11-12</u> Actual Reading				
Impact	Static		132	264	335	454	1040	149	274	470	574	1100
		4070 ft <sup>3</sup> /h <sup>3</sup>	Corrected for Calibration					Corrected for Calibration				
			100	250	400	500	1000	100	250	400	500	1000
105	105	1318	31	15	11			30	15.5	10		
155	156	1602	36	24	15	11		39	21	14	11	
20	20	1820	48	25	17.5	14.5		45	25	17	14	
27	27	2115	50	30	21.5	17.5		48	31	21	16	
365	360	2445	58	32	23	18		56	36	23	18.5	
51	51	2910	73	42	29	26	10.5	73	40	28	22	11
685	685	3340	80	44	35	29	11.5	77	46	33	27	13
91	91	3880		54	41	34	15		55	41	34	16
146	146	4920		77	59	45	18		77	60	43	21
205	206	5830			75	56	24			73	56	26
242	242	6335			83	61	27			80	61	27
299	301	7060				72	32					

FIG. 3. TYPICAL DATA SHEET USED FOR TESTS LISTED IN TABLE 1

In order to determine air entrainment or quantity of air in motion, the average velocity must be obtained. Partial results of about 150 traverses by the grid method (Fig. 4), are given in Table 3.

### Discharge Coefficients

Coefficients of discharge were computed for rectangular orifices with plenum approach. From the results of about 50 tests, shown in Table 3, it appears that this coefficient increases as the aspect ratio increases. Often it is not feasible to use an orifice in practical installations. Therefore in Table 5 are given the values of coefficient of discharge for actual grilles and registers tested. It is noted that this value is approximately 0.8.

*Shape of Stream*

Air streams from the narrow rectangular outlets listed in Table 4 did not retain this shape for more than a few diameters from the outlet. In each case the room air stream became approximately circular. The curves of velocity reduction (Fig. 6), for the rounded-entrance slots were almost identical with the curves of the square nozzles of the same outlet area (see Table 1 for sizes).

The average jet angle for slot orifices as determined by measurements taken at 6 and 9 ft from the outlet, are compared to the average angle

24" x 1" FREE OPENING										VEL. = 500 FPM									
TEST No. 26										DATE 8-15-41									
										DISTANCE - 9 FT.									
										OBSERVERS: GAP & HWE.									
				0	0	0	0	0	0										
			0	30	40	20	10	0	50	0									
			0	60	90	100	90	50	40	0									
			0	100	150	170	160	70	100	30	0								
		0	10	110	150	210	180	100	110	50	0								
		0	20	130	180	250	250	150	90	50	0								
	0	10	10	140	170	210	250	170	80	40	0								
		0	0	170	170	300	290	180	50	20	0								
		0	20	140	140	250	240	150	70	10	0								
		0	50	90	120	170	210	120	50	40	0								
			0	40	70	120	120	120	0	0									
				0	20	50	80	60	0										
					0	0	0	0											

FIG. 4. TYPICAL DATA SHEET FOR STREAM-TRAVERSE TEST

calculated from the entrainment ratio values. Equation (2) based on the momentum theory was used to solve for the angle.

$$\text{Entrainment ratio} = \frac{Q_x - Q_o}{Q_o} = \sqrt{\frac{A_x - A_o}{A_o}} = \sqrt{\frac{A_x}{A_o}} - 1 =$$

$$\left[ \frac{\pi(r_o + X \tan \theta/2)^2}{a_o b_o} \right]^{1/2} - 1 \dots \dots \dots (2)$$

$$\text{where } r_o = \left( \frac{a_o b_o}{\pi} \right)^{1/2}$$

$a_o$  and  $b_o$  = outlet dimensions  
 $X$  = distance from outlet  
 $\theta$  = average jet angle

The angles calculated from the entrainment ratios are almost the same as the observed angles, as shown in Table 6.

*Maximum Residual Velocity for Slots*

Fig. 11 appears to confirm the theory that for a short distance from the outlet the maximum residual velocity for slot outlets varies inversely as the square root of the distance from the outlet. Beyond this region there is a transition period followed by the region where the maximum residual velocity varies inversely as the first power of distance from the outlet.

Experiments indicate that there is a region close to the discharge face for all outlets whether slots, square or round openings in which the maximum



FIG. 5. A FEW OF THE OUTLETS TESTED

residual velocity varies inversely as a fractional power of the distance from the outlet (see Figs. 6 and 11). This relation gradually changes as this distance increases until the velocity varies inversely as the first power of the distance.

#### ANALYSIS AND CORRELATION

*Decrease of Velocity Along Axis*

The theory of the spreading and mixing of turbulent jets has been presented by Tollmein (1926), Bateman (1931), Goldstein (1938),<sup>4</sup> and others. Two

<sup>4</sup>Modern Developments in Fluid Dynamics, British Aeronautical Research Committee, edited by Goldstein, (Oxford University Press, p. 592).

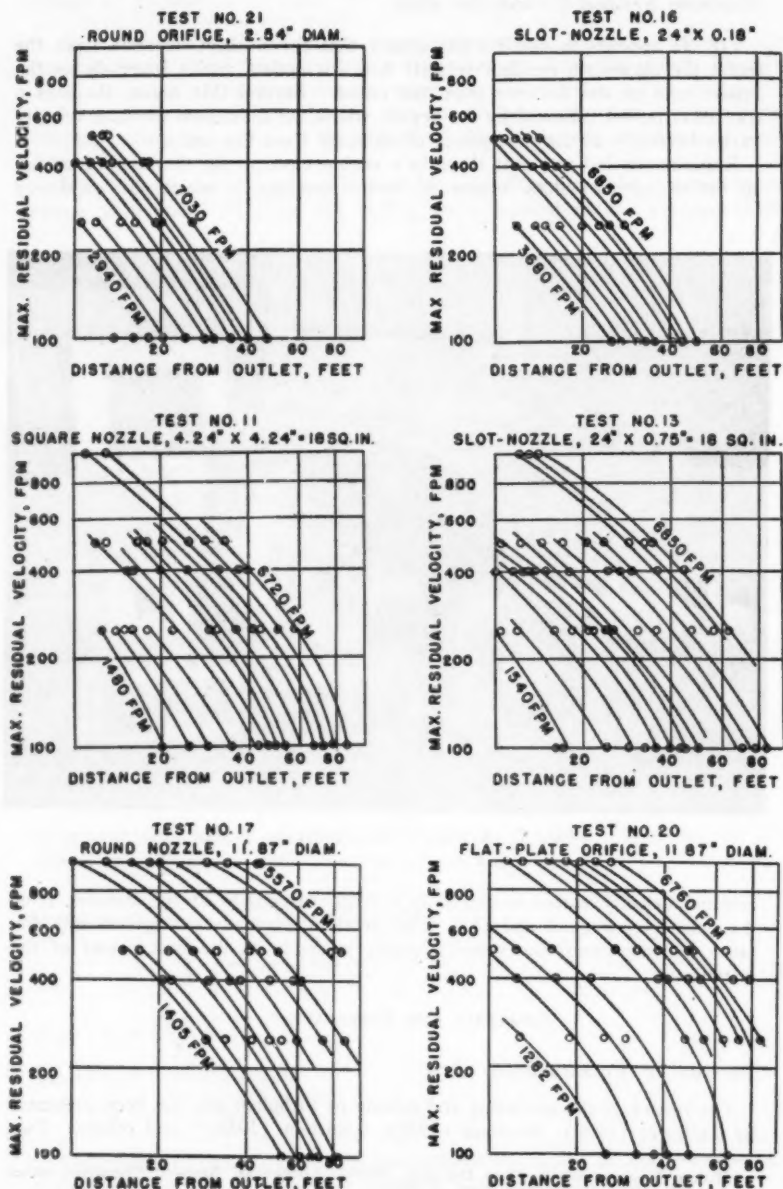


FIG. 6. TYPICAL LOGARITHMIC GRAPHS OF ROOM VELOCITY vs. DISTANCE, ONE CURVE FOR EACH OUTLET VELOCITY

extreme cases are usually treated, (1) jets from symmetrical outlets, and (2) jets from infinite slots. The theory indicates that the maximum velocity along the axis of a symmetrical jet should vary inversely as the distance from the outlet, but that if the stream issues from an infinitely-long slot, the maximum velocity varies inversely as the *square root* of the distance from the outlet. In other words, the velocity-reduction in the case of a stream from a long slot, is less rapid than in the case of a round or square outlet. While this difference was indicated in the experimental results of the present study, the slots used in the tests did not approach an *infinite* length.

Another complication in the behavior of free jets has been well presented by McElroy, in a recent Bureau of Mines Report.<sup>6</sup> Four zones or phases are described by this investigator as the stream progresses from the outlet face: First phase, in which the axis velocity remains almost constant, second phase in which the velocity decreases as the square root of the distance; third phase in which the expansion is free and the velocity decreases as the first power of the distance, and fourth phase in which the velocity decreases more rapidly than before because of return flow or convection currents. From the tests made on the outlets listed herewith, Tables 1 and 4, all of these phases have been definitely recognized. Equation 1 is proposed as applying chiefly to the third phase or zone, extending beyond 25 diameters, but changing gradually after the centerline velocity has dropped below 500 fpm (see Fig. 10). An equation identical with Equation (1) is used by McElroy, and he proposes a *tentative value* of  $K = 6$  for *average centerline velocities*. This is in reasonable agreement with the values recommended in Table 2. Similar values were obtained by Ruden,<sup>8</sup> using a 2.8 in. round nozzle, as is shown by the agreement of curves *A*, *B* and *D*, Fig. 10. Probably some of the reduction of the value of  $K$  (Table 2), is due to the fact that the accidental convection currents in any space interfere with the stream when very low velocities have been reached. Ashley<sup>7</sup> has emphasized the fact that the residual velocity always decreases rapidly near the end of the throw in any practical case, because of the backflow or counter-current set up by the room circulation. McElroy reported that when a stream from a 5 in. round pipe had travelled 25 ft in a square mine drift 8 ft high, the maximum velocity was reduced, due to backflow, to 6 per cent of the initial velocity (Fig. 10, Curve *C*). This corresponds to a value of  $K = 4.0$  (when  $V_0 = 7520$  and  $V_r = 450$ ). It is concluded that the use of Equation (1) and the values of  $K$  in Table 2 are in reasonable agreement with the few data available from other sources.

#### *Ratio of Maximum Velocity to Average Velocity at any Cross Section*

Complete traverses of the cross sections of a stream at great distances from the outlet become almost impossible, because of the number of readings required. While over 200 such traverses have already been made in the course of this investigation, few of these were at distances exceeding 25 diameters, because, even at this distance a single traverse requires at least two hours. The results of about 150 tests indicated that the ratio of maximum

<sup>6</sup> Air Flow at Discharge of Fan Pipe Lines in Mines, by G. E. McElroy, (Bureau of Mines, Report of Investigations—No. 3730, November, 1943).

<sup>8</sup> Ruden, P., Turbulente Ausbreitungsvorgänge im Freistrahle: Naturwissenschaften, Vol. 21, 1933, pages 375-378.

<sup>7</sup> A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 261.

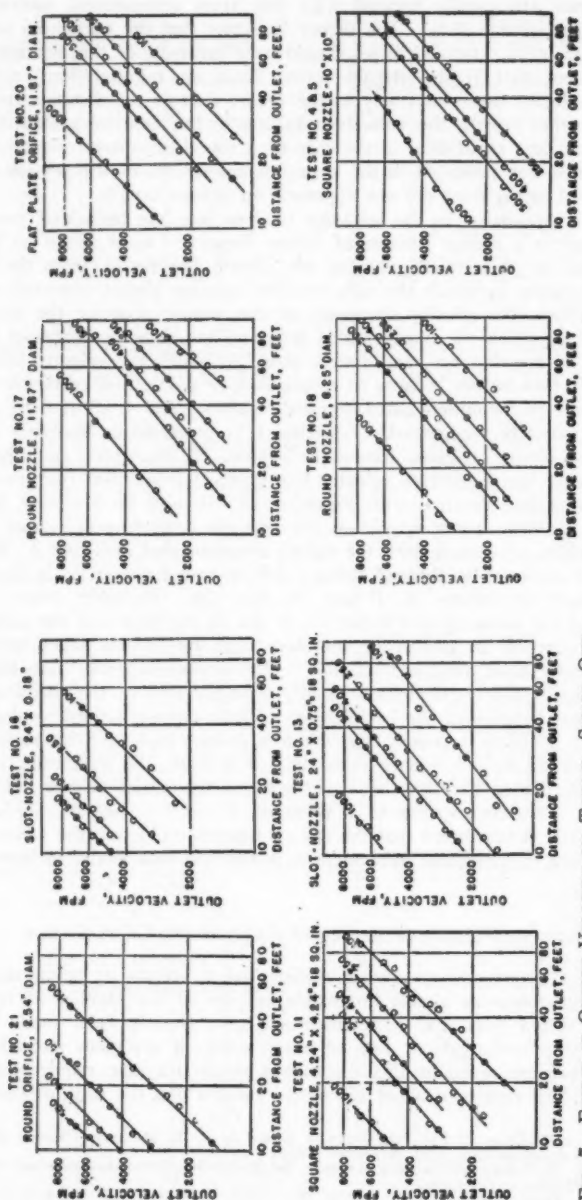


FIG. 7. EFFECT OF OUTLET VELOCITY ON THROW—SMALL OUT-  
LETS. NUMBERS ON CURVES INDICATE MAXIMUM  
RESIDUAL VELOCITY

FIG. 8. EFFECT OF OUTLET VELOCITY ON THROW—LARGE OUTLETS

centerline velocity to average velocity, was around 3 irrespective of outlet size or shape, or of initial air velocity.

In discussing the previous research paper in this series, Coogan and Goff<sup>8</sup> made graphical analyses of the two traverses given by the sample data sheets in this previous paper, using the method described by Förthmann.<sup>9</sup> The

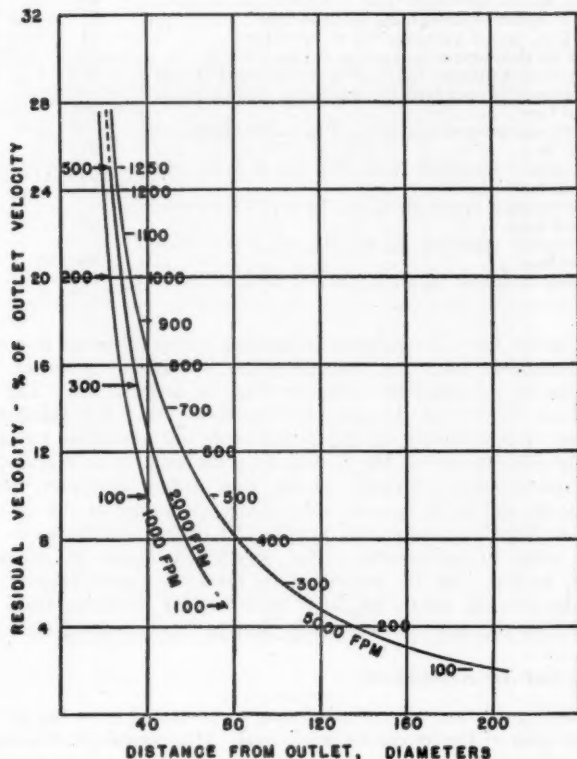


FIG. 9. DIMENSIONLESS PLOT OF RESIDUAL VELOCITY vs. DISTANCE FROM OUTLET; RECTANGULAR COORDINATES.

authors have since plotted many traverses in this manner. McElroy made a similar analysis of his experimental data based on anemometer measurements, and also quoted a value of this ratio,  $R = 2.6$ , from Tollmein's experimental work on a round jet and  $R = 2.4$  from Förthmann's experiments. The A.S.H.V.E. GUIDE<sup>10</sup> states that the maximum velocity is *usually from 2.5 to 3.5 times the average*. However, a careful analysis shows that the value

<sup>8</sup> A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 262.

<sup>9</sup> See N.A.C.A. Tech. Memo. 789, 1936.

<sup>10</sup> HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1943, Chapter 31, Air Distribution, p. 592.

TABLE 5—DISCHARGE COEFFICIENTS FOR REGISTERS AND GRILLES

(Plenum Approach  $\text{cfm}/\text{Gross Area} = 700/(1 \times 0.5) = 1400 \text{ fpm}$ )

DESCRIPTION OF OUTLETS	PER CENT FREE AREA	COEFFICIENT OF DISCHARGE
Grille $\frac{3}{4}$ in. square openings; $\frac{1}{4}$ in. cross bars.....	45	0.870
Grille $\frac{1}{2}$ in. square openings; $\frac{1}{8}$ in. cross bars.....	51	0.788
Register $\frac{1}{4}$ in. square openings; $\frac{1}{8}$ in. cross bars.....	51	0.788
Register 81 vertical openings; approx. 0.3 in. x 1.5 in.....	53	0.799
Register vertical openings; $\frac{1}{2}$ in. $4\frac{1}{2}$ in.; 12 vertical bars.	58	0.770
Register rectangular openings; $\frac{1}{2}$ in., $1\frac{1}{2}$ in.; 10 horizontal, 6 vertical bars.....	59	0.821
Register rectangular openings; $\frac{3}{8}$ in., $1\frac{1}{2}$ in.; 21 horizontal, 6 vertical bars.....	61	0.795
Grille rectangular openings; $\frac{3}{8}$ in., $1\frac{1}{2}$ in.; 27 horizontal, 12 vertical bars.....	62	0.760
Register rectangular openings; $\frac{3}{8}$ in., $1\frac{1}{2}$ in.; 27 horizontal, 12 vertical bars.....	63	0.748
Grille rectangular openings; $\frac{1}{2}$ in., $1\frac{1}{2}$ in.; 5 horizontal, 6 vertical bars.....	66	0.758
Grille vertical openings; $\frac{1}{2}$ in., $4\frac{1}{2}$ in.; 12 vertical bars..	69	0.725

obtained for the ratio of maximum to average velocity depends to some extent on the instrument used for measuring the velocities. A lower minimum velocity can be measured by velometer than by anemometer. The value of  $R=3$  given above was determined from tests using the velometer. The anemometer is satisfactory for measuring centerline velocities, but since it is not usually accurate below 100 to 150 fpm, the total cross section traversed with an anemometer is actually smaller than with a velometer, because the latter may be read to 20 fpm or less. Hence the values of the velocity ratio reported as 3 to 1 by velometer, may be 2.5 to 1 when the same stream is traversed with an anemometer. For practical purposes in evaluating air quantities, mixing and air movement, a ratio of center velocity equal to 3 times the average should be fairly accurate, for distances from 10 to 50 diameters from the outlet. It is probably close to the same for greater distances.

#### Jet Angle and Air Entrainment

The quantity of room air entrained may be computed from the jet velocities if the total area of the jet can be determined. If a conical jet discharges from a round outlet, and the angle of the jet is known, the area at any section is

TABLE 6—CALCULATIONS VERSUS TESTS

(Free Open Slots Plenum Approach  $\text{cfm}/\text{area} = 800 \text{ ft}/\text{min}$ )

OUTLET SIZE IN INCHES	AVERAGE MEASURED JET ANGLE	ENTRAINMENT RATIO AT 9 FEET	CALCULATED ANGLE USING ACTUAL ENTRAINMENT
24 x 3	22.5°	4.62	23.2°
24 x 2	23.0°	6.37	24.0°
24 x 1	23.5°	8.47	24.6°
24 x $\frac{1}{2}$	22.5°	11.85	24.2°
24 x $\frac{1}{4}$	22.5°	17.75	25.4°

of course easily determined, and by multiplying this by the average velocity across the section, the volume-rate of flow is obtained. It has been incorrectly assumed by certain investigators that the jet retained the shape (aspect ratio) of the outlet. Actually, the stream becomes almost symmetrical within a few diameters from the outlet face if the aspect ratio of the outlet is less than 100; hence, for calculating the performance of the stream beyond 10 diameters,

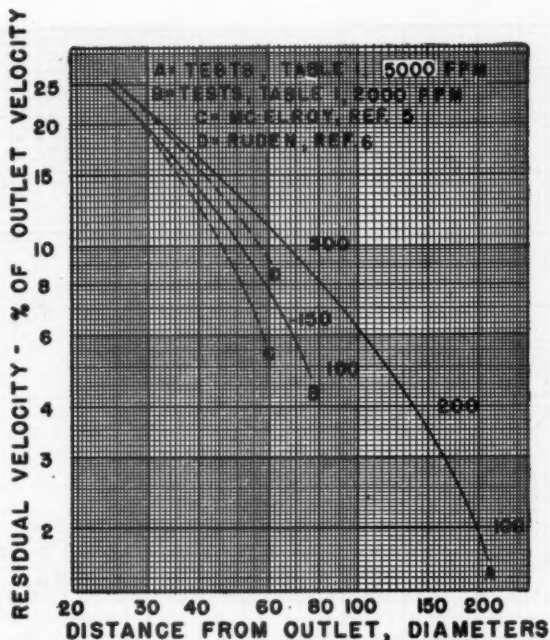


FIG. 10. DIMENSIONLESS PLOT COMPARING RESULTS OF THREE INVESTIGATIONS

it is incorrect to base the calculation on the shape of the outlet. Accepting this fact that there is a short transition length within which the stream is converted to an approximately symmetrical shape, the stream beyond this section may be considered conical, and the entrainment may then be calculated if the jet angle is known, and the apex of the cone is located. Since round and rectangular jets of the same area were found in this investigation to be practically the same beyond 20 diameters from the outlet, it appears logical to assume the apex of the cone is in the same location for any jet, as long as the aspect ratio of the outlet is small. The volume flowing at any section then becomes:

$$Q_x = A_x V_x \dots \dots \dots (3)$$

But assuming a round jet and using Equation (1):

$$V_x = \frac{V_r}{R} = \frac{KV_o\sqrt{A_o}}{RX},$$

$$A_x = 0.785\left(D_o + 2X \tan \frac{\Theta}{2}\right)^2,$$

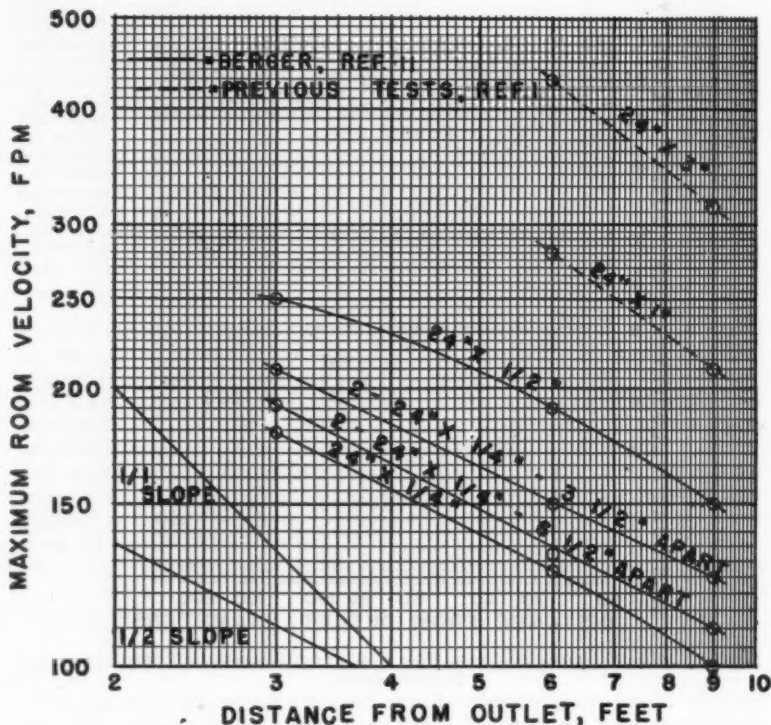


FIG. 11. EFFECT OF SLOT WIDTH AND SPACING ON ENTRAINMENT RATIO

$$Q_x = \frac{0.785KV_o\sqrt{A_o}}{RX} \left(D_o + 2X \tan \frac{\Theta}{2}\right)^2 \quad (4)$$

On the basis of *effective outlet area* as used in Equation 1:  $Q_o = A_o V_o$ . Then the equation for entrainment ratio may be written as follows:

$$\begin{aligned} \frac{Q_x - Q_o}{Q_o} &= \frac{Q_x}{Q_o} - 1 \\ &= \frac{0.785 K}{R X \sqrt{A_o}} \left( \sqrt{\frac{A_o}{0.785}} + 2 X \tan \frac{\Theta}{2} \right)^2 - 1 \quad (5) \end{aligned}$$

Assuming a 20 deg jet angle, this becomes:

Entrainment Ratio =

$$\frac{0.785K}{RX\sqrt{A_o}} \left( \sqrt{\frac{A_o}{0.785}} + 0.35X \right)^2 - 1 \dots \dots \dots (6)$$

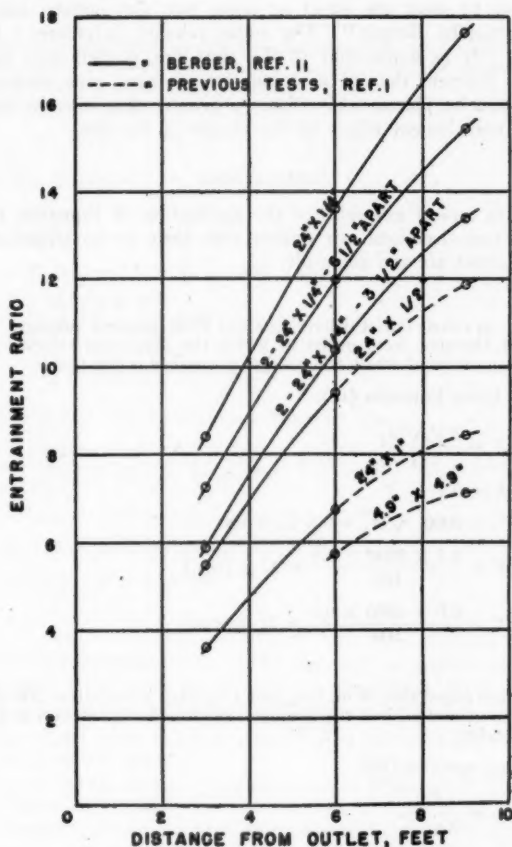


FIG. 12. TYPICAL LOGARITHMIC GRAPH OF ROOM VELOCITY *vs.* DISTANCE, FOR SLOTS; OUTLET VELOCITY 800 FPM, (CFM/AREA)

Two factors that affect entrainment ratio are variation in outlet area and extreme variation in aspect ratio. Fig. 12 shows the comparison of a slot (aspect ratio 24 to 1), and a square opening of the same area. The entrainment ratio is about 18 per cent greater for the 24 to 1 slot. By doubling the aspect ratio of a slot and at the same time dividing the area by two, which

means using a slot of the same length but of one-half the width, the entrainment ratio is increased about 50 per cent. This is shown by comparing slots 24 in. by 1 in., 24 in. by  $\frac{1}{2}$  in., and 24 by  $\frac{1}{4}$  in. (Fig. 12).

#### *Parallel Slots Above One Another*

Figs. 11 and 12 show the effect of using two slot outlets, one above the other, as reported by Berger.<sup>11</sup> The outlet velocity (cfm/area) for Fig. 12 was 800 fpm. It is noted that if the slots are placed close together the effect is little different than if a single slot of equal area were used. The  $\frac{1}{4}$  in. slots must be placed some distance greater than  $6\frac{1}{2}$  in. apart to get the increased entrainment effect of the single  $\frac{1}{4}$  in. slot.

#### APPLICATIONS

Following are typical examples of the application of Equation (1). These apply only to non-directional air outlets with little or no temperature difference between outlet air and room air.

##### *Example 1:*

*Given:* A 6-in. sq nozzle outlet delivering air at 2000 fpm face velocity.

*Required:* (a) Distance from outlet at which the maximum velocity is 100 fpm; (b) Distance from outlet at which the maximum velocity is 500 fpm.

*Solution:* (a) Using Equation (1a):

$$X = \frac{KV_o\sqrt{A_o}}{V_r}$$

From Table 2,  $K = 3.7$

$$V_o = 2000, \sqrt{A_o} = 0.5, V_r = 100.$$

$$X = \frac{3.7 \times 2000 \times 0.5}{100} = 37 \text{ ft (ans.)}$$

$$(b) X = \frac{6.0 \times 2000 \times 0.5}{500} = 12 \text{ ft (ans.)}$$

##### *Example 2:*

*Given:* A square edged slot 18 in. long and 1 in. high is to deliver 300 cfm.

*Required:* The maximum and the average velocities in the stream at a distance of 25 ft from the outlet.

*Solution:* Using equation (1b):

$$V_r = \frac{KQ}{X\sqrt{A_o}}$$

$$A_o = \frac{18 \times 1}{144} \times 0.65 = 0.081$$

(See Table 3).

$$V_o = \frac{300}{0.081} = 3700.$$

$$K = 5 \text{ (estimated).}$$

$$X = 25.$$

<sup>11</sup> Air Stream Characteristics of Narrow Slots, by J. A. Berger. (Thesis, Case School of Applied Science, December 1943, 63 pages.)

$$V_r = \frac{5 \times 300}{25\sqrt{0.081}} = 212.$$

Actual value of  $K$  from Table 2 is 5.2. Hence, maximum velocity at 25 ft distance is  $212 \times \frac{5.2}{5} = 220$  fpm, and average velocity is  $\frac{220}{3} = 73$  fpm.

**Example 3:**

**Given:** A bar-type grille of 65 per cent free area and discharge coefficient  $C = 0.75$  is to be used to produce a throw of 30 ft, with a maximum residual velocity of 150 fpm. Outlet velocity (based on plenum pressure) is 1600 fpm.

**Required:** (a) Area of grille (core area); (b) total room air entrained in the 30-foot throw.

**Solution:**

(a) Using Equation (1a):

$$\sqrt{A_o} = \frac{X V_r}{K V_o}$$

$$X = 30. \quad V_r = 150. \quad V_o = 1600. \quad K = 4.2$$

$$A_o = \left( \frac{30 \times 150}{4.2 \times 1600} \right)^2 = 0.45 \text{ sq ft.}$$

$$\text{Core area} = \frac{0.45}{0.75} = 0.60 \text{ sq ft or } 86 \text{ sq in.}$$

(b) Entrainment ratio is computed from Equation (6):

$$\frac{Q_{30} - Q_o}{Q_o} = \frac{0.785 \times 4.2}{3 \times 30 \times \sqrt{0.45}} \left( \sqrt{\frac{0.45}{0.785}} + 0.35 \times 30 \right)^2 - 1 = 5.92$$

$$Q_o = V_o A_o = 1600 \times 0.45 = 720 \text{ cfm.}$$

Therefore, total room air entrained is:

$$Q_{30} - Q_o = 5.92 \times 720 = 4260 \text{ cfm.}$$

### CONCLUSIONS

The following conclusions seem to be justified, as applied to straight flow outlets up to 100 sq in. area, for outlet velocities below 7000 fpm, when the air stream is not disturbed by contact with surfaces or by adjacent streams:

1. The shape of the outlet has only a small effect on the throw of the stream as long as the aspect ratio is below 50.
2. The shape of the air stream at distances beyond 20 diameters from the outlet is not greatly affected by the shape of the outlet.
3. The performance of an air stream from an outlet depends on the *effective* area of the outlet. Effective area of a rounded-entrance outlet is almost 100 per cent, but in any other case, corrections must be made by multiplying by percentage free area, and by a coefficient of discharge to account for jet contraction.
4. Beyond about 20 diameters from the outlet and as long as the maximum velocity in the stream exceeds 500 fpm, the residual velocity is almost directly proportional to the outlet velocity and inversely proportional to the distance from the outlet.
5. When the maximum residual velocity in the stream no longer exceeds 500 fpm, the velocity falls off more rapidly; hence if the usual methods of calculation are used (Equation 1), the constant of proportionality must be reduced accordingly (Table 2).
6. The numerical values to be used in the equations for air stream performance (Equations 1 to 5) will depend somewhat on the methods and instruments used for measuring the air velocities.

7. The ratio of maximum velocity to average velocity at any section of the stream beyond 10 outlet diameters is approximately 3 to 1.

8. Multiple slots, nozzles or orifices, when spaced only a few inches apart, produce an air stream that performs almost exactly like the stream from a single outlet of the same total area.

#### ACKNOWLEDGMENT

The tests reported in this paper have extended over a period of about two years, and the authors wish to thank the many persons who have assisted in carrying on the work. Special thanks are extended to the following for making large numbers of detailed observations and calculations, and plotting curves: Messrs. K. E. Wolfs, S. M. Fingerhut, S. A. Rosenbluth, J. A. Berger, R. F. Tuve and E. C. Cornell, all of the Class of 1943, Case School of Applied Science.

#### DISCUSSION

C. M. ASHLEY, Syracuse, N. Y. (WRITTEN): Considering outlets in the form of orifices, there is a physical reduction of the size of the stream, but as a matter of fact almost all of the energy available in the form of *static pressure* ahead of the outlet is effective at the discharge. Therefore, I believe that the proper procedure for this type of outlet is to figure the area from the *available static pressure ahead of the outlet*, or from the *total pressure* at the outlets, and the cfm of air, and otherwise to treat the outlet as a nozzle of dimensions smaller than the physical size of the outlet. On the other hand, where bars, grilles, or other similar obstructions are introduced into the outlet, the effect is quite different because the air stream is discharged substantially over the whole face of the outlet and becomes fairly uniform in velocity a few inches or even 1 or 2 inches from the outlet. There is, with this type of outlet, an appreciable loss of available energy, and consequent reduction in velocity due to the conversion from the area of the individual streams to the area of the stream as a whole.

What actually happens, I believe, is that the velocity through the outlet is considerably higher than the equivalent to the static pressure behind the outlet, and that a certain portion of this is converted into pressure directly in front of the outlet. I believe you will find that an appreciable negative pressure is obtained in the streams in front of the outlet and that this pressure is rapidly reduced as the streams converge. However, this conversion process is carried on with considerably less than 100 per cent efficiency so that the resulting final velocity is less than that equivalent to the static pressure behind the outlet. In practice, for an outlet of this sort, I would recommend that the discharge area be based upon the cubic feet of air per minute divided by the measured outlet velocity and that the static pressure be determined independently and applied as an efficiency factor or coefficient of resistance. Alternatively the area of the stream can be measured and the velocity calculated. I believe this problem should be explored further in your continuing program.

AUTHOR'S CLOSURE: Mr. Ashley's comments are appreciated. His experience and ideas will be of practical value to us in planning our future work along this line. We have not attempted, as yet, to do very much investigation of bar type grilles. We wanted to study the fundamental, simple type outlets at the start and then later on introduce temperature differentials and bar type grilles.

We have observed the four phases of the air stream as pointed out by McElroy<sup>12</sup> in his paper. The work that we did was carried on mainly in the third phase. That is the portion of the stream out several diameters from the outlet.

<sup>12</sup> Loc. Cit. Note 5.



**1249**

## DISCOLORATION METHODS OF RATING AIR FILTERS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, University of Minnesota.

ONE of the primary uses to which air filtration has been placed in recent years is for the removal of those air borne dusts which cause discoloration or soiling of decorated surfaces and merchandise. In order that filters may be rated properly as to their ability to reduce such discoloration, it is first necessary to devise satisfactory laboratory test apparatus for the study of efficiencies based upon this principle. However, as with all other problems involving the measurement of air filter efficiencies, the difficulties are legion, to a great extent, because of the lack of strict definition as to the exact property which it is desired to measure. The purposes of the present investigations are:

1. To review the present methods of determining air filter efficiencies by photometric methods.
2. To analyze the phenomenon of discoloration.
3. To develop apparatus for determining the efficiency of air filters based upon a true measure of discoloration.
4. To present comparative test data showing the efficiencies of standard filters rated by different discoloration methods.

### REVIEW OF PRESENT PHOTOMETRIC METHODS OF RATING AIR FILTERS

One of the earlier methods used for rating air filters by photoelectric means was reported by Nutting<sup>1</sup> and consisted essentially in determining the *weight* efficiency of filters, that is, the percentage of the weight of dust removed by a filter. Calibration curves were prepared showing the relationship between the weight of the test dust fed to the filter and the current generated by a photoelectric cell. The dust fed to the filter was collected on a porous filter paper target which was subjected to a light beam on one side and a photometer on the other. With this apparatus, the photoelectric responses for definite weights of dust passing the filter were determined, and it was then possible to calculate the filter efficiency on a weight basis.

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<sup>1</sup> An Alternate Method of Comparing the Dust Arrestance of Air Cleaning Devices, by Arthur Nutting. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 451.)

Presented at the 50th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1944.

The first method of determining filter-efficiency on a true discoloration basis was developed by the Bureau of Standards and reported by Dill.<sup>2</sup> This method consisted essentially in removing air samples simultaneously from up-stream and downstream of a filter, passing these samples through porous filter papers to remove the dust, and then subjecting these papers to a light beam and photometer to determine the concentrations of the dust deposits. In rating filters by this or allied methods, the efficiency may be determined by application of Equation (1):

$$\text{Efficiency} = 1 - \frac{C_1}{C_2} \quad (1)$$

where  $C_1$  = downstream dust concentration.  
 $C_2$  = upstream dust concentration.

The density of dust  $D$ , on the filter paper will vary with the area, the rate of sampling, the concentration, and the time of sampling as in Equation (2):

$$D = \frac{Q_1 C}{A} \quad (2)$$

where  $Q_1$  = volume of air sampled, cubic feet per minute.  
 $t$  = time of sampling, minutes.  
 $A$  = area of dust deposit, square inches.  
 $C$  = concentration of dust in air stream.  
 $D$  = density of dust deposit on filter paper.

If Equation (2) is solved for  $C$ , then,

$$C = \frac{DA}{Q_1 t} \quad (3)$$

This expression may be then substituted in the efficiency equation giving:

$$\text{Efficiency} = 1 - \frac{D_1 A_1 Q_2 t_2}{D_2 A_2 Q_1 t_1} \quad (4)$$

In practice, the application of this relationship requires that three of the four variables,  $A$ ,  $D$ ,  $Q$ , and  $t$ , be kept constant and the other varied. As it is difficult to measure  $D$  in absolute units, but is comparatively easy to determine when  $D_1$  and  $D_2$  are equal, this variable is usually adjusted so that  $D_1$  and  $D_2$  cancel.

In the test apparatus and procedure devised and reported by Dill all tests were made by letting

$$\begin{aligned} D_1 &= D_2 \\ t_1 &= t_2 \\ Q_1 &= Q_2 \end{aligned}$$

and thus,

$$\text{Efficiency} = 1 - \frac{A_1}{A_2} \quad (5)$$

Thus, with this method, the size of the filter papers is varied by providing different apertures corresponding to the different efficiencies. In order to determine when the densities of dust on the two filter papers are equal, these papers are subjected to a light beam on one side and a photometer on the

<sup>2</sup> A Test Method for Air Filters, by R. S. Dill. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 379.)

other. It is then considered that the dust densities are the same when the amounts of light transmitted through the filter papers and measured by the photometer are equal.

A modification of this method was devised by Rowley and Jordan<sup>3</sup> in which,

$$\begin{aligned} D_1 &= D_2 \\ I_1 &= I_2 \\ A_1 &= A_2 \end{aligned}$$

and thus,

$$\text{Efficiency} = 1 - \frac{Q_2}{Q_1} \dots \dots \dots (6)$$

In this modification, the volumes of air sampled on the two sides of the filter were varied until the resulting dust spot densities on the porous filter papers were equal. In addition, a further modification was adopted for determining the relative discoloration of the filter papers by subjecting them to a beam of light and determining by means of a photometer the amount of light reflected. Both the incident and the reflected light beams were at angles of 45 deg to the filter paper surface and 90 deg to each other. Reflected light measurements were used in place of transmitted light measurements because this more nearly simulates the mechanism of the human eye in evaluating discoloration.

All three of the methods here reviewed for rating the efficiency of air filters by photometric means result in different efficiencies when applied to identical filters. The reasons for these filter rating discrepancies are fundamental as each method measures a somewhat different property of the filter and there has, as yet, been no attempt at strict definition as to the exact measurement desired. The desirability appears evident, therefore, for an analysis of the phenomenon of discoloration and the methods by which such discoloration can best be determined.

#### THEORY OF DISCOLORATION

Soiling or discoloration is a surface phenomenon dependent for its detection upon a source of incident light and its subsequent diffuse reflection. The brightness and wave length of the reflected light depend upon the color and nature of the surface. The degree to which a soiled area is evident is dependent to a great extent upon the contrast between the soiled area and the surrounding unsoiled areas. If the soiling is even, it may not be readily recognizable as such even though the actual degree of soiling may be greater than in another case of similar soiling made more evident by contrast with surrounding undisclored surfaces.

The discoloration of surfaces can take place by several other means than by soiling brought about by dust deposits. For example, discoloration by fading as a result of weathering or exposure to sunlight is common. However, as such means of discoloration are not allied to the present problem of air filtration, all references to discoloration in this article will infer discoloration by dust only.

In studying the problems involved in the measurement of discoloration, it is necessary to introduce several terms which are frequently used ambiguously.

<sup>3</sup> A.S.H.V.E. RESEARCH REPORT NO. 1169—A Comparison of the Weight Particle Count and Discoloration Method of Testing Air Filters, by Frank B. Rowley and Richard C. Jordan. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 29.)

However, these terms have definite meanings in the field of photometry, and those meanings will be adopted for present purposes:

**Lightness** (reflectance) is the ability of a surface to reflect light diffusely. Lightness is frequently expressed as percentage reflection or reflectance with reference to a standard surface assumed to reflect diffusely 100 per cent of incident light. A pure magnesium oxide surface is commonly used for reference because of its exceedingly high reflective characteristics. However, for strictly comparative purposes, any light colored surface may be used.

**Brightness** is the actual amount of light reflected from a surface; thus, the brightness of reflected light is dependent not only upon the nature of the surface but

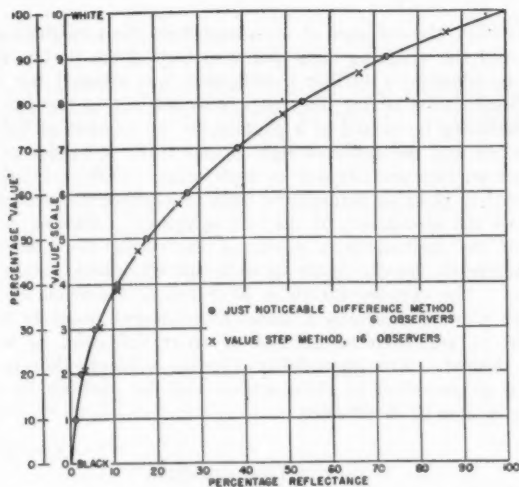


FIG. 1. RELATIONSHIP BETWEEN PHYSICAL (REFLECTANCE) AND PSYCHOLOGICAL (VALUE) INTERPRETATIONS OF THE LIGHTNESS OF A SURFACE<sup>4</sup>

also upon the intensity of the incident light beam. Therefore, a surface of high lightness will possess zero brightness when no light beam is incident. Brightness is commonly expressed in foot candles.

**Value** is the psychological and physiological counterpart of lightness. There are no common units of value, but they usually increase numerically from black to white and must be chosen so as to be representative of equal change increments in appearance. Fig. 1 presents the value scale determined by Munsell, Sloan, and Godlove.<sup>4</sup> This figure shows the relationship between value and percentage reflectance as determined by using the *just noticeable difference* and the *value step* methods.

In analyzing Fig. 1, it is interesting to note that in passing from black to white on the value scale, the initial psychological changes are much greater than the initial physical or percentage reflectance changes, and that the final psychological changes are much less numerically than the final physical changes. Applying this to surface discoloration or soiling, the initial discoloration of a light surface by a dark dust will not be as evident as an equivalent initial

<sup>4</sup> Neutral Value Scales I. Munsell Neutral Value Scale, by A. E. O. Munsell, L. L. Sloan, and I. H. Godlove. (*Journal of the Optical Society of America*, Vol. 23, p. 394, Nov. 1933.)

discoloration of a dark surface by a light colored dust. In this respect it should also be noted that, although we normally think of soiling in terms of the discoloration of a light surface by means of a dark dust, this is not necessarily so. A dark colored surface may be just as readily soiled by a light colored dust, and in fact, such soiling is more readily discernible by the human eye. However, as most dusts are not truly light in color, such discoloration of dark surfaces is not so commonly experienced.

The degree of discoloration of a surface may be measured either by visual or photoelectric means. Visual comparisons while simple are inaccurate because



FIG. 2. ASSEMBLED VIEW OF TEST APPARATUS

they introduce the human element. On the other hand, trained observers are not required if photoelectric means of reflectance measurement are introduced and furthermore, the sensitivity of the human eye may be reproduced at different wave lengths by the application of a properly designed light filter. For these reasons the several methods of evaluating the discoloration efficiency of air filters developed during the past few years have, practically without exception, used photoelectric means for measurement of light.

The most direct approach to the measurement of air filter efficiencies by discoloration methods is to actually discolor either light or dark surfaces by means of a contrastingly colored dust fed into the air. Thus, the discoloration of white surfaces may be made by black or brown dusts and the discoloration of black surfaces by white or gray dusts. The problem may be further complicated by the introduction of intermediately colored surfaces discolored by contrasting dusts. In such cases, however, it would be possible

to have a visual discoloration as evidenced by actual differences in color composition, and yet have the same percentage reflectance of light from both surfaces. Such an analysis of discoloration would necessitate the introduction of a *whiteness* concept involving analysis of the spectrum composition of the reflected light. In most cases such a procedure would needlessly complicate the efficiency determinations and be of little additional value.

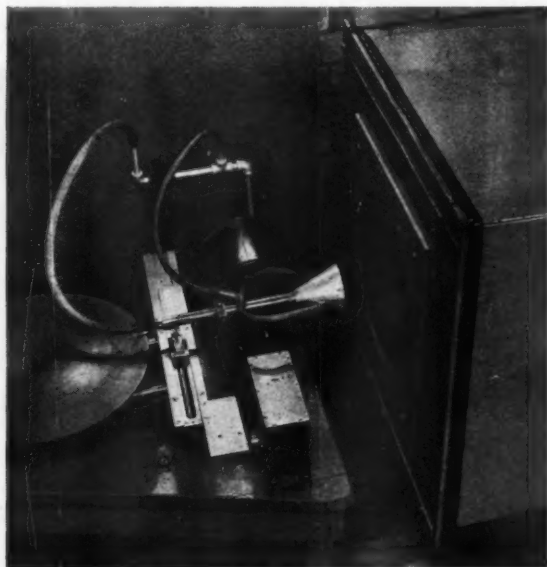


FIG. 3. VIEW OF MODIFIED DUST FEEDER

#### DESIGN OF APPARATUS FOR MEASURING DISCOLORATION EFFICIENCIES

In order to approach as closely as possible a true measurement of the discoloration efficiency of air filters and also to permit modifications of these efficiencies based upon the physiological and psychological reactions of the human eye to discoloration, some changes were made in the apparatus and procedure formerly used at the University of Minnesota for rating air filters.<sup>6</sup> The test duct used for introducing dust to the filter and for measuring the air volumes and the static differential across the filter was identical with that formerly used and is shown in Fig. 2. The dust feeder shown in Fig. 3 was modified in order that small amounts of dust might be accurately weighed and fed uniformly into the air stream. An aluminum channel 6 in. x  $\frac{1}{4}$  in. x  $\frac{1}{4}$  in. is used to hold the dust. This channel is moved at a uniform velocity past a pickup tube having an inlet the same width as the channel. This pickup tube is connected to the low pressure section of a small Venturi tube through which compressed air is passed. The velocity of the entering air picks the ribbon

<sup>6</sup> Loc. Cit. Note 3.

of dust from the channel and carries it through the tube to a distributing nozzle at the entrance of the filter test apparatus. This apparatus is essentially the same as that previously used for making weight efficiency determinations of air filters<sup>6</sup> with the exception that in the former case the dust was picked up from a flat revolving disk.

The sampling tube used for drawing the air-dust sample from the duct for purposes of discoloring the filter papers is identical with that previously used

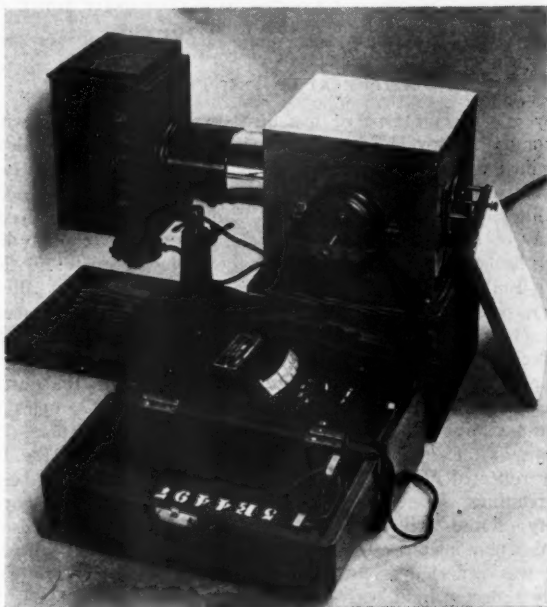


FIG. 4. PHOTOMETER FOR MEASUREMENT OF DISCOLORATION

in the University of Minnesota photometric efficiency tests.<sup>7</sup> However, in this case, instead of using two sampling tubes, one placed upstream and one downstream of the filter, a single sampling tube located at the bell orifice is used. This sampling tube is shown at the right of Fig. 2. The dust samples are drawn from the test duct through the sampling tube by means of a suction pump, and the volumes of the samples measured by means of a calibrated flat plate orifice located in the sampling tube.

In order to measure the discoloration of the filter papers by means of diffusely reflected light, the apparatus shown in Fig. 4 was used. This consists of a hollow cube 6 in. x 6 in. x 6 in. coated on the inside with several layers of a magnesium oxide paint. On one side of the cube is a hole of 1 in.

<sup>6</sup> A.S.H.V.E. RESEARCH REPORT No. 1094—Air Filter Performance As Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity Through Filter, by F. B. Rowley and R. C. Jordan. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 415.)

<sup>7</sup> Loc. Cit. Note 3.

diameter through which a parallel beam of light is allowed to pass. Directly opposite this opening is another of  $1\frac{1}{4}$  in. diameter against which the filter paper to be measured is placed. On the third wall of the cube and at right angles to the beam of light passing through the cube, a third opening of  $1\frac{1}{4}$  in. diameter is located. A photoelectric cell is placed at this opening so that any light reflected within the cube from the filter paper will affect the cell. The photonic cell used is equipped with a filter so that its sensitivity to different light wave lengths is approximately the same as that of the human eye. This photoelectric cell is connected to a photometer which enables the measurement of the reflected light in foot-candles.

#### DEVELOPMENT OF TEST PROCEDURE

With any one specific type of test dust used in conjunction with any one specific color of background filter paper, it is first necessary to determine the amount of dust to be fed before maximum discoloration is reached. In the present tests two colors of background were used: one, the normal white uncolored filter papers, and the other, similar filter papers dyed black by means of a jet black ink. Tests to determine the maximum discoloration against such backgrounds were made with the following dusts: air-floated silica (200 mesh), Fuller's earth (200 mesh), grain dust (200 mesh), Illinois fly-ash (200 mesh), powdered rosin (100 mesh), Cottrell ash (200 mesh), Pocahontas ash (two different samples taken from different seams—200 mesh), and lampblack (100 mesh).

In making these maximum discoloration determinations, the dusts were fed into the test duct without any filter in place, and the samples drawn out through the sampling tube and deposited upon the filter papers. The actual discolorations of these papers were determined by measuring the foot-candles of diffusely reflected light using the apparatus previously described. All discoloration readings were, of course, relative, as they depended not only upon the intensity of the light reflected from their surfaces, but also upon the design of the photometer. In all cases, the apparatus was adjusted so that when light was reflected from a white, blank filter paper, the brightness of the diffusely reflected light was 40 foot-candles. When the black, blank filter paper was substituted for the white one, the brightness reading was 7 foot-candles. This reading was not zero for two reasons: first, all of the light incident upon the black filter paper was not absorbed, a small percentage of it being reflected diffusely in the photometer cube; and second, some stray rays from the light source were incident upon the walls of the reflecting cube adjacent to the location of the black filter paper. The results of these maximum discoloration tests are shown photographically in Fig. 5. It should be kept in mind that no black and white photograph is capable of accurate reproductions of the visual appearance of these papers, with the possible exceptions of the discolorations obtained by means of air-floated silica and lampblack. With the other dusts there was a considerable range in color not possible to reproduce.

Theoretically, the maximum discoloration using any one type of test dust should be the same regardless of the color of the background. Actually, however, the maximum discoloration was limited to a certain extent by the ability of the dust to cling to the background surface. In most cases, it was found that the dust deposits would tend to flake off the surface of the

filter paper before a deposit sufficiently thick to obscure completely the color of the background could be made. It is for this reason that there are discrepancies between the maximum discolorations for a specific type of dust when deposited against both a black and a white background. For example, the maximum discoloration of air-floated silica deposited on a black background proved to be 35 foot-candles, whereas the same dust deposited against a white background resulted in a reading of 37 foot-candles.

These preliminary tests indicated the weights of dust which must be fed to the air stream to result in maximum discoloration of the filter papers when no air filters were located in the test apparatus. They also indicated which

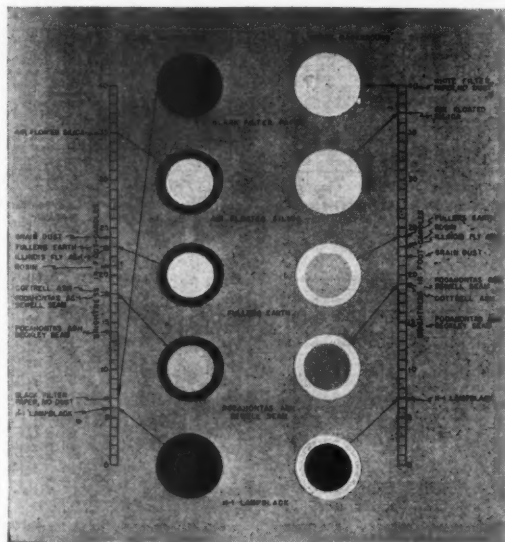


FIG. 5. MAXIMUM DISCOLORATION OF BLACK AND WHITE FILTER PAPERS BY VARIOUS TEST DUSTS

test dusts used in conjunction with either a black or a white background would be most practical for filter discoloration efficiency tests. For example, it would be obviously impractical to use lampblack in conjunction with a black background as the diffusely reflected light is approximately the same with or without the lampblack dust deposit. On the other hand, there is a discoloration range of from 40 foot-candles to 7 foot-candles or a spread of 33 foot-candles between lampblack deposits on white filter paper and the plain white background paper itself. It is evident that the greatest contrast between an undisclored background and a discolored background is desirable from the standpoint of increasing the sensitivity of the efficiency tests. From this standpoint the two most desirable combinations investigated were lampblack used in conjunction with a white background and air-floated silica used in conjunction with a black background.

In establishing an efficiency versus light reflectance scale, 100 per cent efficiency was taken as corresponding to the light reflectance from the blank filter paper with no dust deposits. A definite weight of dust was then chosen to be fed to the filter under test, this weight being somewhat less than that causing maximum discoloration of the filter paper and also less than that giving heavy enough deposits on the filter paper to cause flaking. When this weight of dust was fed into the test duct with no filter in place, the light reflectance corresponding to the resulting dust deposit on the paper was taken

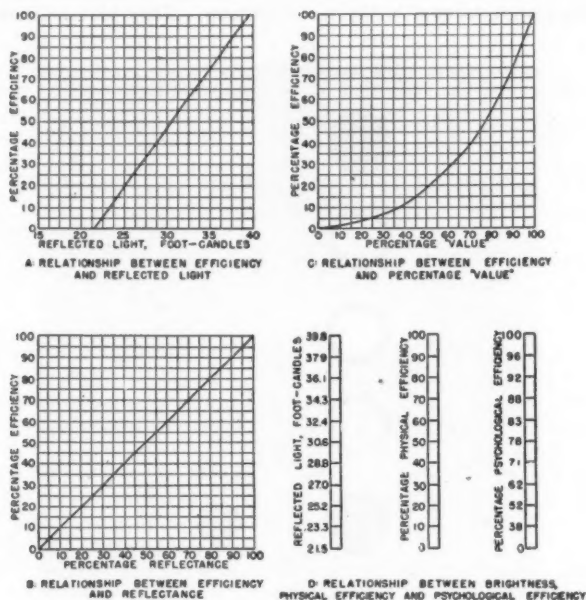


FIG. 6. RELATIONSHIPS BETWEEN BRIGHTNESS, PHYSICAL DISCOLORATION EFFICIENCY, AND PSYCHOLOGICAL DISCOLORATION EFFICIENCY FOR LAMPBLACK AGAINST WHITE BACKGROUND

as that for 0 per cent efficiency. These points of 100 per cent and 0 per cent efficiency were chosen to correspond to the conditions in which none of the dust and all of the dust fed to the filter passed through the test apparatus. It was then assumed that the relationship between filter efficiency and diffusely reflected light was linear between these two limiting points. In this way the filter efficiency was made directly dependent upon the degree of the resulting discoloration as determined photometrically. Thus, the efficiency is a function of the degree of discoloration alone with no definite relationship between filter efficiency and weight of dust fed or filter efficiency and dust particle count.

Figs. 6 and 7 show the evolution of the relationships between brightness of diffusely reflected light, physical discoloration efficiency, and psychological

discoloration efficiency. Fig. 6 is for a dark dust, lampblack, on a white filter paper background. In these figures *Curve A* shows the relationship between brightness of reflected light and filter efficiency as determined photometrically. *Curve B* shows the relationship between percentage reflectance and filter efficiency. In the case of lampblack on white filter paper, 0 per cent reflectance was taken to correspond to the foot-candles of reflected light at 0 per cent efficiency and 100 per cent reflectance to correspond to the foot-candles of reflected light at 100 per cent efficiency. In the case of the light dust against a

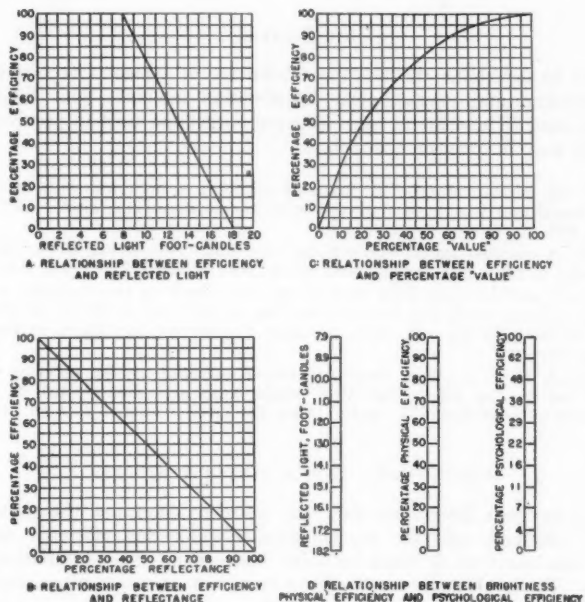


FIG. 7. RELATIONSHIPS BETWEEN BRIGHTNESS, PHYSICAL DISCOLORATION EFFICIENCY, AND PSYCHOLOGICAL DISCOLORATION EFFICIENCY FOR POWDERED SILICA AGAINST BLACK BACKGROUND

black background, 100 per cent reflectance corresponded to 0 per cent efficiency, and 0 per cent reflectance to 100 per cent efficiency. *Curve C* of both figures shows the relationship between percentage physical efficiency as determined by the photometer and percentage psychological efficiency corresponding to the reactions of the human eye. These curves were taken from the data presented in *Curve B* and the relationship between percentage *value* and percentage reflectance shown in Fig. 1. In this manner the discoloration efficiency is presented in terms of a scale based upon the visual conception of discoloration. It is related in no direct way to the physical quantity of dust which has passed through the filter but is based entirely upon the human eye's conception of the potential discoloring properties of the air-dust mixtures passing the filter. From an inspection of *Curve C*, it is evident that the human eye is

less sensitive to slight discolorations of light surfaces by means of dark dusts than are photometric measuring devices. On the other hand, the human eye is more sensitive to slight discolorations of dark surfaces by means of light dusts than are photometric measuring devices.

*Curves D* of Figs. 6 and 7 present by means of bar diagrams a direct comparison between the brightness of the reflected light in foot-candles, the percentage physical efficiency, and the percentage psychological efficiency or *value*. These bar diagrams present in another form a direct comparison of relationships shown in *Curves A, B* and *C*.

#### AIR FILTERS

In order to determine the relative efficiencies of typical filters when rated by the physical and psychological discoloration methods, four filters were chosen for test. These filters were identical with those used in previous tests,<sup>a</sup> and briefly may be described as follows:

*Filter A*—A permanent type of cleanable oil filter 4 in. thick with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

*Filter B*—A viscous coated throw-away type filter two inches thick, the fibrous media graded in fiber size, density, and oiling from entering to leaving side.

*Filter C*—A cellular type filter two inches thick built in two sections, each with the axis of cells set at 45 deg to the center line of duct and at 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side of the filter.

*Filter D*—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion-pleated in frame to give an area of approximately twelve times the cross sectional area of air stream.

#### DISCOLORATION EFFICIENCY TEST RESULTS

Each of the four filters described was tested to determine the physical discoloration efficiency and the psychological discoloration efficiency using six different combinations of black or white background with different colors of test dust. Three of these combinations consisted of white filter papers with lampblack, Pocahontas ash, and powdered rosin as test dusts, and the other three with black filter papers using silica, Pocahontas ash, and powdered rosin as test dusts. The Pocahontas ash and rosin were both intermediately colored dusts and, therefore, were applicable to either a black or a white background. The air-floated silica was very white and, therefore, could only be used against a black background, and the lampblack could only be used against a white background. Strictly speaking, the relationship of Fig. 1 used in deriving the psychological discoloration efficiencies is applicable only to the cases of the black dust on a white background or the white dust on a black background. Wide application of colored dusts with this method of testing would require the establishing of an accurate reflectance versus *value* curve for the particular type of dust used.

The results of these tests, made at a filter air face velocity of 300 fpm, are presented in Table 1. The first column of this table records the color of the background filter paper, the second column the type of dust fed, and the third

<sup>a</sup> Loc. Cit. Note 6.

TABLE 1—COMPARISON OF PHYSICAL AND PSYCHOLOGICAL DISCOLORATION EFFICIENCIES

TYPE OF DUST FEED	COLOR BACK- GROUND PAPER	WEIGHT DUST FEED, GRAMS	FILTER A		FILTER B		FILTER C		FILTER D	
			Physical Discoloration Efficiency	Psychological Discoloration Efficiency	Physical Discoloration Efficiency	Psychological Discoloration Efficiency	Physical Discoloration Efficiency	Psychological Discoloration Efficiency	Physical Discoloration Efficiency	Psychological Discoloration Efficiency
Lampblack.....	White	1.0	34	66	32	64	7	30	52	78
Air floated silica.	Black	10.0	47	19	58	28	23	8	57	27
Pocahontas.....	White	6.0	37	68	26	58	28	60	64	85
Pocahontas.....	Black	8.0	70	38	74	42	45	18	80	48
Rosin.....	White	8.0	..	..	30	62	41	72	32	64
Rosin.....	Black	8.0	67	34	73	41	34	13	61	30

TABLE 2—COMPARISON OF EFFICIENCIES DETERMINED ON FOUR FILTERS BY EIGHT DIFFERENT TEST METHODS

METHOD OF DETERMINING EFFICIENCY	EFFICIENCY FILTER A		EFFICIENCY FILTER B		EFFICIENCY FILTER C		EFFICIENCY FILTER D	
	Pocahontas Ash	Lampblack	Pocahontas Ash	Lampblack	Pocahontas Ash	Lampblack	Pocahontas Ash	Lampblack
Weight.....	93	45	96	63	85	33	90	84
Particle count.....	..	..	33	..	22	..	40	..
Optical density, transmitted light.....	25	10	36	33	18	10	48	71
Optical density, reflected light.....	51	37	48	54	30	9	67	71
Physical discoloration, white background.....	37	34	26	32	28	7	64	52
Psychological discoloration, white background.....	68	66	59	64	61	30	85	78
Physical discoloration, black background.....	70	..	74	..	45	..	80	..
Psychological discoloration, black background.....	36	..	40	..	19	..	46	..

column the weight of dust fed. The dust weights were somewhat under those which, when tested with a filter of 0 per cent efficiency, would result in a dust deposit susceptible to flaking from the surface. The next eight columns of the table present the test results for filters A, B, C, and D. It was found impossible to determine a valid rating for filter A using rosin dust against a white background, as it was found that the color of the dust changed in passing through the filter which resulted in fictitious negative efficiencies. This apparently was caused by a reaction between the rosin dust and the filter oil which resulted in a blacker shade of dust than the original entering the filter. This may likewise have had some effect upon the other efficiency ratings reported for filter A when using the same dust with a black background, and possibly even for the other three filters rated. However, as there was no evidence to substantiate this possibility, the remaining results have been recorded for comparative purposes.

As was to be expected, in all cases where dark dusts were used against white background filter papers, the psychological discoloration efficiencies were considerably higher than the physical discoloration efficiencies. As may be seen by an inspection of the relationships shown in Fig. 6, the increase in psychological over physical efficiency is the greatest when the physical discoloration efficiency is the lowest and the least when the physical discoloration efficiency is the highest. The greatest and the smallest percentage changes in efficiency in translating from the physical to the psychological basis were noted in these tests for lampblack tested on filter C with a white background paper in which the efficiency increased from 7 per cent to 30 per cent, and for Pocahontas ash tested on filter D with a white background paper in which the efficiency increased from 64 per cent to 85 per cent. In the first case, the increase in efficiency was 329 per cent and in the second case, 33 per cent.

As indicated by the relationships shown in Fig. 7, the psychological efficiency will always be lower than the physical efficiency when black background filter papers are used in conjunction with light colored test dusts. In these cases, the smallest percentage changes are found on translating physical discoloration efficiencies to psychological discoloration efficiencies when the initial physical discoloration efficiencies are highest. The maximum percentage changes are found when the initial physical discoloration efficiencies are the lowest. However, the range in percentage changes for the different filters tested was not as great in the case of the black background filter papers as in the case of the light background filter papers, the maximum being 65 per cent and the minimum 40 per cent.

Table 2 presents a comparison of the efficiency tests for filters A, B, C, and D rated by eight different methods using Pocahontas ash and five different methods using lampblack. The weight or crucible method referred to its essentially that described in the A.S.H.V.E. Code.<sup>9</sup> In determining efficiencies by this method, a known weight of dust is injected into the air upstream of the filter, and samples of the air and dust are removed downstream of the filter and weighed. The weights of dusts entering and leaving the filter are used to determine the efficiency. The particle count method referred to consists essentially of removing air samples simultaneously upstream and downstream of the filter and determining by microscopic examination the number of particles per unit volume of air. The filter efficiency is based upon the rela-

<sup>9</sup>A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225.)

tive particle count concentrations upstream and downstream of the filter.<sup>10</sup> The optical density method of rating air filters referred to in Table 2 is that previously described in this paper in which the filter efficiency determinations are made by the application of Equations (1) and (6). For purpose of comparison, tests have been made by this method both with photometric measurements made using transmitted light and using reflected light. The physical and psychological discoloration efficiencies referred to in Table 2 are those made by the filter rating methods presented in this paper and compared in Table 1.

These efficiency determinations summarized in Table 2 for the two types of test dusts and the four different filters indicate clearly the extremely wide variations in test results which may be obtained using a single filter with a single type of test dust. In all cases with any one filter using a single type of test dust, the weight efficiencies are the highest, and with the exception of filter C, the particle count efficiencies are the lowest. Although the six remaining methods of determining filter efficiency are all some form of discoloration test method, the divergence in results on any one filter with a single test dust is great. For example, the results on filter C with Pocahontas ash range from 18 per cent to 61 per cent, and those for filter A with Pocahontas ash range from 25 per cent to 70 per cent. Such divergent results are principally the result of a lack of strict definition as to the property to be measured. Strictly speaking, the psychological discoloration efficiencies are probably the closest approach to the inherent reduction in discoloration available in a filter as judged by the human eye. Even with the psychological discoloration efficiencies on any one specific filter using a specific test dust, there is a range of results available dependent upon the color of the background upon which this dust is deposited. However, it would appear more logical to utilize a white background for use with a black or nearly black dust as this appears to simulate the commonest source of discoloration.

### SUMMARY

In this paper the principal photometric methods used at the present time in rating air filters have been reviewed. It has been shown that no two of these methods result in the same air filter efficiency ratings when applied to identical filters using the same type of test dust, principally because each method measured a somewhat different property of the filter.

In an attempt to arrive at a rational basis founded solely upon the ability of the filter to reduce the potential discoloring properties of an air-dust mixture, the theory of discoloration has been analyzed. It has been shown that discoloration as measured by photometric means alone is not sufficient to fully define discoloration as judged by the human eye. For this reason the term *value* has been introduced in order to take into consideration the psychological and physiological reactions of the human eye in interpreting discoloration.

A new method of determining the true discoloration efficiency of air filters has been developed. This method is based upon the assumption that a filter of 100 per cent efficiency will filter the air to a point where samples of the air-dust mixture passing the filter and drawn through porous filter papers will

<sup>10</sup> Loc. Cit. Note 3.

cause no discoloration, and the air-dust samples from a filter of 0 per cent efficiency will cause discoloration equivalent to that which would be obtained were no filter present. All intermediate stages of discoloration are assumed to be in direct linear relationship with filter efficiency. Thus, by this method the degree of discoloration of filter papers is taken as the sole criterion of efficiency, and there is no direct relationship between this discoloration efficiency and the removal of dust as measured on a weight or particle count basis. These physical discoloration efficiencies have been translated by a *value*—reflectance relationship into what has been termed a *psychological discoloration efficiency*, thus taking into consideration the human eye's interpretation of discoloration.

Four typical filters were chosen and tests made upon them to determine both the physical and psychological discoloration efficiencies. These tests were made using both porous white filter papers and porous black filter papers for the backgrounds to be discolored and four test dusts as the discoloring media. These dusts were air-floated silica, Pocahontas ash, powdered rosin, and lampblack. In addition, comparisons are reported between all four filters showing the efficiency ratings obtained by eight different methods using Pocahontas ash and six different methods using lampblack.

No specific recommendation is made as a result of this research toward the adoption of any one method of rating the discoloration efficiency of air filters. The results are reported primarily to augment the collective knowledge of the field and to point the direction toward a possible final solution. It is realized that although the method of attacking this problem appears logical, it is, at the same time, somewhat radical, and that further research would be required before general adoption of such a method could result.

## DISCUSSION

G. W. PENNEY, East Pittsburgh, Pa. (WRITTEN): I think that it is important to look into all means of measuring efficiency of air cleaning devices. This paper confirms my opinion that the discoloration test substantially as described by Dill in 1938 and referred to by Rowley as a *matching efficiency* is the most useful test that has been proposed.

The information which I would like about any air cleaning device would consist of a determination of efficiency as a function of both particle size and type of material to be removed. I believe that such a test is obviously impractical for anything except a very extensive research on a particular device. I believe that no simple test can measure adequately all of the functions needed in an air cleaning device. We must then choose a test that comes the nearest to measuring the desired factors and is sufficiently simple to conduct. The discoloration test proposed by Dill measures the relative time required to produce a given discoloration which, I believe, is the factor of greatest interest in most applications.

The *psychological efficiency* as described in this paper has several objections. It is supposed to give the relative soiling as estimated by the eye after a given period of operation with and without an air cleaning device. However, in practice we do not want to clean walls, draperies, etc., at definite periods when we install an air cleaning device. We are instead more interested in lengthening the periods between cleanings. If I interpret the paper correctly, it seems to me that the efficiencies given do not correctly interpret the values given for the response of the eye in Fig. 1. The tests given use only a partially blackened sample, and yet is applied to the curve of Fig. 1 as though this sample were entirely black. This exaggerates the discrepancy between the various methods of measuring efficiency. The exact definitions of effi-

ciency and methods of calculating efficiency are hard to follow as given, and I believe it would help if in the discussion the authors would give sample calculations.

In conclusion, this examination of various methods of measuring efficiencies has served to confirm my opinion that the method substantially as given by Mr. Dill is the most satisfactory method of determining a discoloration efficiency since it is simple and closely approximates the factor in which we are usually most interested.

PROFESSOR ROWLEY: In analyzing and discussing this paper, it should be kept in mind that this presentation is made not to recommend any changes in the present methods of testing and rating of air filters but merely to augment the collective knowledge of the field. Certainly the present methods of testing filters are not entirely closed to question, and with this in mind we thought it advisable to review the fundamental premises upon which air filter ratings are based. The somewhat radical method of rating filters by discoloration included here with the other more commonly used methods is certainly not perfected. All current methods of rating which are satisfactory have been subjected to years of developmental work. The authors fully realize that there are many minor flaws in test procedure which would have to be changed if the method here introduced were ever adopted for limited use in a specific phase of the air filter problem.

In analyzing Fig. 1 it should be realized that the value scale shown is only relative, with arbitrary representation of black and white. The points marked 0 and 10 representing the blackest and the whitest surfaces cannot be considered to have 100 per cent absorption and 100 per cent reflection of light energy. The points between these two extremes represent partially blackened surfaces. For all practical purposes this is analogous to the conditions under which the present tests were conducted. The extremes of black and white as shown in Fig. 5 are those approached in the curves representing the actual rating of the filter. It is true that for practical purposes the heaviest deposits used on any of the filter papers were limited by actual sealing of the dust from the papers. This condition, however, is approached only under conditions close to complete discoloration. There is probably a small error introduced because of this lack of complete discoloration, but this in no way affects the basic theory back of the test, and this was the primary consideration.

As to Mr. Penney's question, which of the discoloration methods of test should be used, consideration should be given to the property which is being measured. Are we trying to measure discoloration as we see it on draperies, walls, etc., or are we trying to measure the physical efficiency of the filter using a discoloration method merely as a measuring stick? These are two different things, and I think the question as to what we want the filter to do and what we are trying to measure needs clarifying. This has been one of the primary difficulties in establishing an air filter test code.

It cannot be brought out too forcefully that the authors would be the last to desire complete substitution of the method of test described here for the one originally developed by Mr. Dill. We, ourselves, continually use a modification of Mr. Dill's original test and have no intention of substituting for it the one suggested here. We do, however, strongly believe that the currently used discoloration test methods do not fully present the picture of filter performance.

R. S. DILL, Washington, D. C. (WRITTEN): This paper is regarded as a useful addition to our fund of knowledge since it is evident that the test method and the optical concepts described have received consideration and trial. However, the paper and its results are not expected to have a material influence on air filter testing because simpler effective test methods are known and because it is doubtful whether many people will accept the theory that air cleaner effectiveness can be based on ability to reduce visual discoloration in the manner described. No relation has been established between the ability of an air cleaner to prevent discoloration of a filter paper through which a portion of its effluent air has been drawn and the ability of the cleaner to prevent dirtiness in a room ventilated with air from it. Presumably, acceptable criteria at present, so far as effectiveness is concerned, are that the filter

is best (1) which will arrest most dust of a given size or (2) which will arrest the finer dust.

Air cleaner tests with filter paper and reflected light were tried at the National Bureau of Standards in 1934 and 1935 for the reason that photometers suitable for this purpose were conveniently available. Such tests were abandoned in favor of transmitted-light tests for the reason that results were easier to reproduce with transmitted light. When transmitted light is used, the effect obtained is largely independent of dust color. When the deposit density is not too great, each dust particle interrupts its own fraction of the light and hence registers its proportionate effect.

It is unfortunate that the paper under consideration does not give information on reproducibility of results. It would be interesting to know whether the data presented are average and whether or not it was necessary to apply the method of least squares in order to arrive at the figures shown. Of these figures, those in Table 2 for the Weight Test Method and for the Transmitted-Light Test Method are considered to be of most interest. The great difference in the efficiencies indicated by the two methods is noted and we surmise that it is attributable to the great size range of the test dusts used. In the test dust there were probably a few large particles which the filters were very effective in arresting, which accounts for the high efficiency by the weight method, while there probably were many small particles, which the filters were not so effective in arresting, which accounts for the comparatively low efficiency by the transmitted-light method. For an absolutely homogeneous dust, if such were attainable, efficiencies by the two methods would be expected to coincide, within the experimental error. The desirability of a dust of small size range is considered to be indicated for efficiency tests.

Nothing is said in this paper about filter paper selection. At the *National Bureau of Standards*, it has been found necessary to use pairs of filter papers for each test which are optically matched before use. Otherwise, acceptable reproducibility of results is not achieved. Such matching is necessary even though filter papers are cut for use from the same sheet.

No sufficient reason has appeared for not using a sampling tube both upstream and downstream of a filter under test. This method is applicable to tests of electrostatic equipment with air-borne dust as well as to other cleaners with dust injected into the air. The method utilizes the photometer as a *null* instrument and assumptions about the relation between efficiency and transmittance or between efficiency and reflectance are avoided.

It is probable that continued research on test dusts, on the dust holding capacity of filters and on the actual effectiveness of air cleaners in keeping houses cleaner and more healthful is in order. I, for one, would be content to let the so-called blackness or discoloration test remain substantially as it has been developed and is now in use.

PROFESSOR ROWLEY: Of course it is impossible to answer all of Mr. Dill's questions because the question of filter testing is such a broad one and there are so many variable factors involved. Insofar as selecting the filter paper is concerned, we found practically no variation between surface reflection from filter papers of a good quality chosen from a single lot. However, the same filter papers that showed little difference in surface reflection would undoubtedly show great differences in transmitted light if subjected to a light beam on one side and a photoelectric cell on the other. This was one of the practical disadvantages which we found in using transmitted light for filter discoloration ratings, as apparently a good quality of filter paper would still have great variations in density and thickness, thus making it very difficult to match papers properly before starting tests involving transmitted light. Thus, our findings are in complete agreement with Mr. Dill's insofar as selecting filter papers to be used for transmitted light tests are concerned. However, in any of our tests involving reflected light, we found this difficulty to be practically eliminated.

I am not sure that I understand Mr. Dill correctly, but insofar as rating a filter by the discoloration method and having the results agree with what that filter would

do in practice, I think the results depend upon the type of dust and the type of background used for the rating. It is true that no definite relationship has been established between the ability of an air cleaner to prevent discoloration of a filter paper through which a portion of effluent air has been drawn and the ability of the cleaner to prevent dirtiness in a room ventilated with air filtered by the cleaner. However, the same criticism applies equally well to any method of air filter testing used at the present time, with differences only in the property being rated. Of course, a filter is rated under one set of conditions, and is used under all combinations of conditions. Such a rating is, therefore, not going to hold for all practical applications of the filter.

MR. DILL: Is that difficulty you mention applicable also to the weight method?

PROFESSOR ROWLEY: If you use the weight method as we used it in the Standard Code, then I don't think it is. The difficulty there is related to the size of dust particles in the test dust. One of the greatest obstacles, I believe, has been to find a test dust which can be duplicated accurately and which would simulate the dust to which the filter is to be subjected in practical operation.

J. W. MAY, St. Matthews, Ky. (WRITTEN): The authors should be commended for their new and interesting approach to the problem of discoloration methods of rating air filters. The general subject of discoloration efficiency has many ramifications and a paper of this nature not only adds to the collective knowledge of the subject but it stimulates interest and thought in a field which has been somewhat neglected in the past.

In reviewing the present photometric method for rating air filters, I would like to ask the authors why they feel that a more desirable means for determining discoloration efficiency can be made by varying the volumes of the air sampled on the upstream and downstream side of the filter in preference to varying the area of the target.

As early as 1934 the *National Bureau of Standards* reported discoloration tests in which both the air volume and the area were varied, but there has always been some doubt in my mind as to whether or not different rates of sampling on the upstream and downstream side would involve catching more or less dust than should actually be represented by the sample. In other words, is it not desirable to maintain the same velocity in the sampling tube as in the duct? It appears that equal velocities would give more reasonable assurance of catching the pro-rata amount of dust. Aside from this there is always the difficulty of accurately measuring small volumes of air.

In Fig. 1 the authors show a very interesting curve giving the relationship between the physical and psychological interpretations of the lightness of a surface. I do not know whether or not this curve was plotted from data going in one direction only, but it is assumed that it will hold equally well for values from black to white as from white to black. The authors mention, however, that strictly speaking Fig. 1 is applicable only to the cases of black dust on a white background or white dust on a black background and is perhaps not too reliable for intermediate values. This raises the question as to whether the two original backgrounds as used by the authors would also apply without some reservation, because Fig. 5 indicates that lampblack on a black background gives a lower foot candle reading than the background alone.

Fig. 5 also indicates that with a white background the foot candle reading ranged from 40 ft-candles with the background alone to 7 ft-candles with the lampblack spot. A similar range of from 7 ft-candles to 35 ft-candles was read when using the black background with and without air floated silica. When Fig. 6A was plotted for lampblack on a white background, only the range of 40 ft-candles to 21.5 ft-candles was covered, with the 21.5 ft-candle value representing 0 per cent efficiency, and the relationship between filter efficiency and reflected light was assumed to be linear.

Fig. 7A showing air-floated silica on a black background also shows a range considerably less than is indicated by Fig. 5.

The authors did not explain why a reduced quantity of dust was fed during these calibration tests, and no mention was made as to the permissible upper limit of target dirtiness or density. However, some experimental work of a similar nature was performed in 1939 in a laboratory with which I am familiar, and it was found that the relationship between reflected light and percentage efficiency was not a straight line unless particular care was taken not to overload the target. As a white background becomes covered with dust the reflected light is progressively reduced, but after the background is rather thoroughly covered, additional layers of dust will have only slight effect upon the reflectance. As a matter of fact, results of these tests indicated that for the dusts tested the relationship between dust fed and light absorbed was not linear when the absorption exceeded approximately 22 per cent.

This is a point which merits consideration, especially if crucible and discoloration tests are to be made simultaneously. Unless the concentration is reduced to a very low value, it will be necessary to limit the length of time for making the discoloration test in order to prevent overloading of the target.

The one other point which I would like to raise is that if black background targets are used, curve 7C indicates that a physical discoloration efficiency of 95 per cent would indicate a percentage *Value* of only 73 per cent. This means that a slight error in determining the physical efficiency in the range of from 95 per cent to 100 per cent would make a considerable difference in the *Value* readings. Conversely if a white background were used, the *Value* percentage would never fall below 72 per cent unless the physical discoloration efficiency of that filter were 40 per cent or less. This indicates that in most cases the useful portion of the *Value* curve is considerably reduced, irrespective of the type of background that is used for the target, and therefore a slight difference in psychological efficiencies might represent a considerable spread in physical discoloration efficiency.

**PROFESSOR ROWLEY:** On the question of background, it is true that if you have anything affecting a straight line relationship, then on one end or the other of that curve, you are going to have an efficiency variation which may be exaggerated by some change on the other ordinate. That is true of the curves under discussion because it could not be otherwise if you vary from a straight line.

The permissible upper limit of target dirtiness was determined by making several preliminary tests to determine the maximum loading of the filter paper. It is true that if a white background becomes colored with dust, the reflected light is progressively reduced, but after the background is rather thoroughly covered, additional dust deposits will have only a slight effect on the reflection. However, in the present tests the upper and lower limits alone were established and a linear relationship assumed between these two in order to define basically the discoloration efficiency. Thus, the efficiency was based upon the degree of discoloration of the surface and not upon the amount of dust required to cause that discoloration.

I should like to point out that the filter paper method of measuring the dust in the air was the same method used many years ago in the Society's Research Laboratory.<sup>11</sup> This was known as the Anderson-Armspach Method in which filter papers were used to determine the actual weight of the dust in the air, assuming the resistance to flow of air through the paper as a measure of the amount of dust collected on the paper.

When you put dust on a filter paper, there are several ways of measuring it. You can put it on a scale and weigh it, you can measure its discoloration by transmitted or reflected light, or you can measure the resistance to air flow through the filter paper. The method used in the Society's Laboratory 15 to 20 years ago in which the resistance to passage of air through the filter paper was taken as a

<sup>11</sup> A.S.H.V.E. RESEARCH REPORT No. 631—A New Method of Making Air Dust Determinations, by F. Paul Anderson and O. W. Armspach. (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 221.)

measure of the weight of dust collected was merely another way to determine the amount of dust collected on the paper.

The question as to the desirability of determining discoloration efficiencies by varying the volume of air sampled on the upstream and downstream sides of these filter papers in preference to varying the area of the target was thoroughly discussed in a previous article.<sup>12</sup> In this paper the results of tests made using varying air volumes and differential air velocities was discussed.

G. W. HEWITT,<sup>13</sup> East Pittsburgh, Pa. (WRITTEN): In this paper, the authors present two new types of efficiency which make use of discoloration—the *physical* efficiency and the *psychological* efficiency. These efficiencies are based on entirely different considerations than those on which the earlier type of discoloration efficiency was based. This earlier type, which may be called the *quantitative* or *matching* efficiency, is described in this paper also. These three types of discoloration efficiency vary widely, and the question arises as to which one has the most logical basis. Two of them are based on the degree of discoloration of surfaces after being subjected to a given quantity of cleaned and uncleaned air; the third is based on the amounts of cleaned and uncleaned air required to produce a given degree of discoloration (*i.e.*, the *matching* of discoloration). The way in which air cleaners are actually used, determines which of the three is the most logical efficiency rating.

Suppose that an air cleaner is to be used in the following way. The walls, draperies, etc., in the cleaned air space are to be cleaned at definite intervals of time regardless of the degree of soiling (*i.e.*, independent of the air cleaner efficiency). In this case, the degree of discoloration which the surfaces reach in the fixed interval between cleanings, is a function of the efficiency of the cleaner. In this case, the physical and psychological efficiencies have more meaning than the matching efficiency. Which of the two is the more significant depends upon the relative importance of the reflectance of the soiled surface, and the human eye's interpretation of the degree of soiling.

Next, suppose the air cleaner is used in a different way. The walls, draperies, etc., are cleaned not at regular intervals, but only when they reach a certain degree of soiling. The efficiency of the air cleaner then affects not the maximum degree of soiling, but the *length of time between cleanings of walls, draperies, etc.* In this case, is not the quantitative or matching efficiency the most logical since this efficiency rating is based on equal degrees of discoloration, or *matching*?

Besides this question as to which is the most logical method, I have two other questions. One has to do with the effect of degree of discoloration of the filter papers (the discoloration obtained on unfiltered air may be designated the *base* discoloration). Is the indicated efficiency, either physical or psychological, independent of the choice of this *base* discoloration?

The second question also has to do with *base* discoloration. In Fig. 1 *value* is plotted from *black* to *white*. In the efficiency test the *base* sample is not *black* but *gray*. In calculating the psychological efficiency using Fig. 1, is the *gray base* assumed to correspond to the *black* end of the value scale? A change in zero is involved here. In the particular case shown by the authors in Fig. 6, the *base* reflectance appears to be approximately 44 per cent, and yet this apparently was assumed to correspond to black in Fig. 1. Taking this change in zero into account, I have calculated psychological efficiencies much lower than those shown in the bar diagram of Fig. 6.

The curves in Fig. A show the various efficiencies plotted as functions of the reflected light readings, for the particular case shown in Fig. 6. Curves A and B are the physical and psychological efficiencies respectively, given in Fig. 6 by Professors Rowley and Jordan. Curve C shows the psychological efficiency which I have calculated from the same data, taking into account the change in zero from a *black* to a *gray base* sample. The discrepancies between Curves B and C, as shown

<sup>12</sup> Loc. Cit. Note 3.

<sup>13</sup> Westinghouse Research Laboratories.

on the slide, are large. It seems to me that the methods of calculating the efficiencies need clarification to eliminate such uncertainties.

Curve D shows the quantitative or matching efficiency as calculated by use of a curve from an earlier paper<sup>14</sup> by Professors Rowley and Jordan, showing the relation between reflectance and the quantity of carbon dust on the filter paper. It is realized that this curve was taken with a different type of reflectance measuring equipment, so it may not be applicable here. For this and other reasons, Curve D is shown on this slide not as an accurate representation of matching efficiency but simply as some indication of its probable relation to the other types of efficiency.

Matching efficiency has been used by several investigators and reported in several papers. It was referred to by Penney<sup>15</sup> in 1937, described by Dill<sup>16</sup> in 1938, and by Rowley and Jordan<sup>17</sup> in 1941. If, in general, air cleaners are used more for the

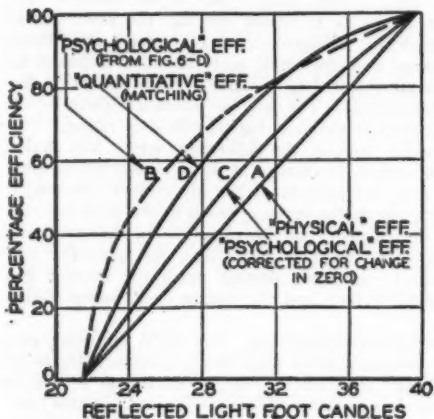


FIG. A—COMPARISON OF VARIOUS TYPES OF EFFICIENCY

- A—Physical Eff.—from Rowley and Jordan's present paper.
- B—Psychological Eff.—from present paper, not corrected for zero shift.
- C—Psychological Eff.—from present paper, but corrected for change in zero from a black to a gray base sample.
- D—Matching Eff.—approximate only, using data from an earlier paper (1).

purpose of increasing the time intervals between cleaning of walls, draperies, etc., rather than for decreasing their apparent discoloration at the end of fixed intervals of time, then the matching efficiency would appear to be the significant one.

PROFESSOR ROWLEY: I might say that in applying the scale of Fig. 1 to any test, the lower reflectance of the test values was taken at the black end of the scale and the higher reflectance was taken at the upper end. In this way the scale was applied directly, although as mentioned in the paper, with some error when dust and surfaces other than black and white were used.

There is another factor concerned with determinations of matching efficiency which has not as yet been discussed. Matching efficiency will not determine the efficiency of a filter if subjected simultaneously to more than one kind of dust and more than one dust particle size. This is so because some filters take out one kind of dust more easily than other kinds. For example, if Pocahontas ash and lampblack are mixed together, some filters will eliminate Pocahontas ash very effectively but not lampblack. However, the lampblack is the component having the greatest effect upon the matching efficiency, whereas the filter has actually taken up a high percentage of the

<sup>14</sup> Loc. Cit. Note 3.

<sup>15</sup> A New Electrostatic Precipitator, by G. W. Penney. (A.I.E.E. Transactions, Vol. 56, 1937, p. 159.)

<sup>16</sup> Loc. Cit. Note 2.

<sup>17</sup> Loc. Cit. Note 13.

Pocahontas dust. Thus, if filters are rated by the matching method, they cannot be applied in practice with any expectation that the rating will be valid. It may, therefore, be seen that almost any method assumed or selected for rating filters will result in some difficulties when applied practically to field ratings.

Y. S. TOULOUKIAN, West Lafayette, Ind.: It seems the standard method and the method just discussed have in turn their advantages and drawbacks. The main item that was stressed in this discussion was the cleanliness of the housewife's curtains and draperies. Its importance is admitted but one other important thing was definitely neglected: the material damage and deterioration caused by dust, besides its effect on cleanliness of appearance. Both of these methods unfortunately do not consider this factor. The damage caused by dust to books in our large libraries is a good example of what I mean.

Speaking just from the aesthetic point of view again, the manner in which curtains become dirty is entirely different from the way a filter test sample becomes dirty in our experiments. Therefore, this method of discoloration of filter papers will not show, at all, how curtains do get dirty. The process is entirely different.

To my mind this paper brings out a psychological point: it shows whether white shows easier on black background or black shows easier on white background.

W. H. CARRIER, Syracuse, N. Y.: I would like to ask the authors if consideration was given to a method that would test precipitrons, which remove fine particles of dust, fumes, and so forth. I know that for example you can breathe cigarette smoke on one side of your filters and see smoke on the other side.

W. L. FLEISHER, New York, N. Y.: I think one of the points that was brought out in the written discussion and has not been stressed sufficiently, is the time element in rating filters. In the electrical photometric method of comparison, we find the blackness test is set up as an indication of the efficiency of the filters but that these blackness tests are taken for only a 10-min period at, of course, an accelerated rate of dust fed. It is my opinion that this method is not in any way accurate. Discoloration is peculiar and cumulative, and tests in which blackness is a measure should be run until the various filter interceptors show approximately the same blackness, which, in my opinion, they all would do after a period which is well within the time that filters would be efficient for use.

PROFESSOR ROWLEY: In regard to Dr. Carrier's question, we did not rate anything insofar as the fine dusts which would be eliminated by an electrical precipitator are concerned. There is, however, no reason why this type of test could not be applied to the rating of an electrical precipitator unit by discoloration.

Mr. Fleisher has made a suggestion that if we could run tests over a period of time comparable to those in practice, we might get something that was more nearly applicable to practical filtering. These tests were, of course, all accelerated as it is obviously necessary to do in order to get any kind of laboratory rating in a reasonable period of time. However, the question of various rates of dust feed and length of test has been thoroughly discussed in a previous paper.<sup>18</sup>

C. E. BENTLEY, San Francisco, Calif.: I think the purpose of filters might be thought of from the health viewpoint, more than from the aesthetic viewpoint. What is the size of the particle necessary to carry contagious diseases, and has any filter test been made regarding that?

PROFESSOR ROWLEY: Our tests have all been made from the physical and not the medical standpoint. Considerable work has been done medically as to the size of

<sup>18</sup> A.S.H.V.E. RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by Frank B. Rowley and Richard C. Jordan. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 339.)

particles which will affect the lungs, etc., but I believe that from a physical rating standpoint, we were not justified in entering this phase of air filtration. Furthermore, all evidence up to the present time indicates that the nature and concentrations of dust particles as normally experienced in everyday life are not particularly detrimental from the health standpoint when considering the dust alone.

MR. FLEISHER: As to the question of bacterial filtration—I am very much interested that this question has come up at this time, because I have stressed it before the Society for the last five or six years. The method I proposed is the incubation method of discovering the size of particle which would carry bacteria or coli. In a way, it is a very simple method to pursue and has been carried out by other people besides myself and by many bacteriologists. I have often carried out this type of experiment for myself. If you take an ordinary atmospheric type of dust and then take samples before your filter and samples after your filter, it is perfectly simple to incubate Petrie dishes and find out whether or not a development of colonies occurs on both slides. You will find that on the entering Petrie dishes in any air, any air from any place, no matter how clear or fresh or pure you may think it is, there is incubation of a large variety of colonies of different kinds of coli, bacteria, and mold spores. A great many of the simpler filters, not electric precipitators, will show almost complete absence of the development of these bacteria and coli beyond the filters.

You can measure pretty well microscopically the size of particle which will penetrate or pass through the filter, and tests would indicate that no particles under 3 microns, which are fairly easily filtered out, will carry the ordinary coli or bacteria. I do not say they may not carry virus.

This is a method that I suggested some years ago for the Society to take up as a method of rating filters from a health standpoint; and, as it is one of the simplest methods I know, I prescribe it as a desirable research project for the Society.

H. B. NOTTAGE, East Hartford, Conn.: I have a comment concerning the general technique of comparing the response of the eye to the response of a photoelectric cell. It should be recognized that the eye and the normal photoelectric cell do not respond to the same relative extent to irradiation of different wave lengths. Further, both individual photoelectric cells and individual eyes may have different relative responses, one to the other among themselves.

Now, it is possible to filter a photoelectric cell so that its response very closely approximates that of the *average* eye. Thus, if discoloration rating methods represent comparisons on the basis of the net reflected intensity of a specified initial illumination, then it seems that there should be obtained a more comparable response between the eye and the photoelectric cell. However, the eye-rating scale presumably includes something other than a measure of total reflected intensity; this would be the physiological property known as *contrast sensitivity*. Thus, the eye would also react to the contrast between black specks and a white background, which may largely explain the relative shift between the two scales.

PROFESSOR ROWLEY: The ordinary photoelectric cell is considerably more sensitive than the eye in the long and the short wave length regions of the physical spectrum, and, furthermore, its sensitivity extends into the infra-red and ultra-violet regions. In order to determine what effect this had on discoloration efficiencies, tests have been made and previously reported in which determinations were made with a correction filter applied to the photonic cell in order to duplicate the spectral sensitivity of the human eye. There were no noticeable differences in the results obtained with and without this correction filter during tests made by the reflected light discoloration methods. These tests were reported in a previously published article<sup>19</sup> to which reference has already been made.

<sup>19</sup> Loc. Cit. Note 3.



**1250**

## THE AXIAL FLOW FAN AND ITS PLACE IN VENTILATION

By W. R. HEATH\* AND A. E. CRIQUI,\*\* BUFFALO, N. Y.

THE FIELD of ventilation is finding a newcomer in its midst, in the axial flow fan. Competing as it does with the centrifugal fan in its use for pressure applications, the fan users' first question is, *When should I use an axial flow fan?* It is the purpose of this paper to point out the characteristics of the axial flow fan and to compare them with centrifugal fans, to provide a basis for answering this question.

### DEFINITION

Broadly speaking, the term *axial flow* is generic, indicating a fan in which the air flows parallel to the axis of the wheel, as opposed to radial flow as in the centrifugal fan. The term might thus be generally applied to desk fans, propeller fans, disk fans, etc., even up to turbine rotors. Among the engineering profession, however, it has come to mean specifically that type of axial flow fan of recent design which unfortunately bears no distinguishing name but which incorporates airfoil section blades, large hubs and general streamlining suitable for operation against pressure. Such a wheel is illustrated in Fig. 1. Actually, any axial flow fan when creating pressure might more accurately be called a *spiral flow* fan since the air cannot both enter and leave the rotor in a true axial direction. It is only when fitted with guide vanes at the inlet or diffuser vanes at the discharge that the air flow through the fan is truly axial.

The advantages of an axial flow type of fan have long been recognized—the prime advantage being the inherent ability to handle large volumes of air in a small space and with a straight-through blow, thus frequently eliminating awkward and space using duct formations. The broad field of application of propeller fans and disk fans bears witness to this. These fans were usually limited in application to those installations where the resistance to flow was low and comparatively stable. Variations in pressure were apt to cause wide variations in the volume of air handled and, in many designs, an increase in resistance was accompanied by a large increase in horsepower consumption. In addition, efficiencies were low, being materially below those prevalent with centrifugal fans. When operating at free delivery, these low efficiencies were not too serious since the power consumption was small anyway for the volume of air being handled, and good efficiencies at free delivery with centrifugal fans were also difficult to obtain without using excessive sizes. When, however, these

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types of fans were used against even moderate resistances, power consumption was usually much higher than with reasonable sizes of centrifugal fans.

#### DESIGN

Axial flow fan development had its origin in the widespread research of the aeronautical sciences, which has been intense throughout the world in the last two decades. As a logical complement to aerodynamic experimentation, airplane propellers were adapted as prime movers for the air in wind tunnels and fan development along these lines followed in many places. The beginning

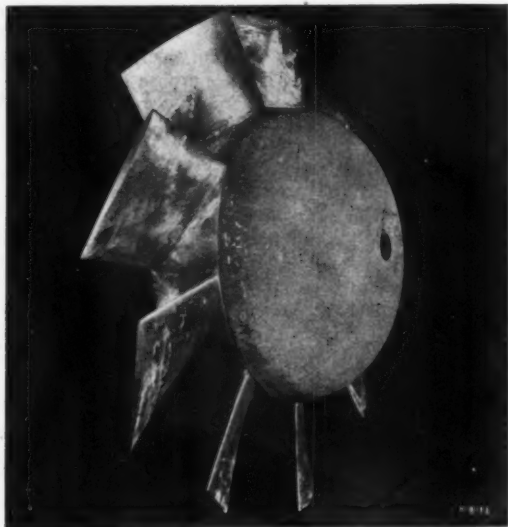


FIG. 1. AXIAL FLOW WHEEL

of this development was in Europe and it is interesting to note that as long as 15 years ago commercial fans bearing all the major features of the present day fans were available in Europe. Figs. 2 and 3 show such fans as illustrated in English catalogs published in 1931 (note the streamlining, large hubs, twisted blades and airfoil vanes).

With the abundant supply of data on airfoil sections available through aeronautical research, it was possible to apply the knowledge to a fan wheel for the purpose of creating airflow just as it was applied to a propeller, with the desire to produce air thrust. The axial flow fan blade roughly follows the appearance of a propeller in that a cross-section of either blade has the familiar airfoil shape and the blade is twisted to greater angles as the center of either type rotor is approached. Applying aerodynamic principles to the design of an axial flow fan shows that the propeller type blade with practically no hub at all is fundamentally unable to produce high pressures efficiently. Because the

linear velocity of the blade near the center of the wheel is so low compared to the tip velocity, it is not practical to build such a wheel to operate at low relative air velocities. At points on the fan curve away from free delivery, the portion of the blades near the wheel center can do little work, so they are simply eliminated and replaced by hub. Thus it is found that present wheel designs have hubs of 50 per cent, 60 per cent, and sometimes as high as 70 per cent of the wheel diameter.

The combination of twisted blade and large hub thus gives substantially constant flow over all the working face of the wheel, which is essential if high efficiencies are to be attained when working with high pressure differences.

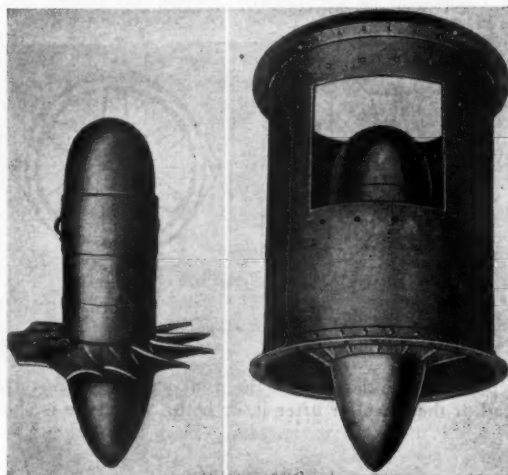


FIG. 2. EARLY ENGLISH FAN

For centrifugal fans, a shallow, wide fan blade indicates a low pressure, high capacity fan. Conversely, a long narrow blade, as in a supercharger, is used for low capacity high pressure conditions. Analogously, small hub axial flow fans are for low pressure, high capacity ratings, while large hub wheels are for high pressure applications.

These characteristic large hubs permit the introduction of the motor in the case of a direct-driven unit, or the bearings, in a belt-driven style, inside the fan without obstructing the air flow. For reasons explained later, the axial flow fan is an inherently high specific speed fan and its outlet velocities are usually higher than with centrifugal fans. The result is that the axial flow fan frequently has a diameter no greater than the outlet of a centrifugal fan. Fig. 4 illustrates the comparative sizes of an axial flow and a centrifugal fan used for the same conditions.

Needless to say, best efficiencies demand streamlining of all parts and careful manufacture with close tolerances. For fans drawing from the room with no duct work on the inlet, belled entrance pieces are usually indicated. For direct

connected units, the normal large hub permits locating the motor close to the wheel without obstruction. In special cases, the optimum hub diameter must be exceeded to meet the practical problems of design presented when the motor is larger than normal.

Another axial flow fundamental is that when pressure is produced in a wheel, the air necessarily leaves in a spiral motion. Fig. 5 shows the velocity vector diagram across an axial flow blade section.

From this diagram and description, the inherent spiral nature of the air flow leaving the wheel is evident; also evident is that deviation from a true axial direction is more pronounced as the volume handled is decreased. To correct this spiral motion, stationary guide vanes are used. These recover much of the rotational component of the velocity leaving the wheel and in so doing

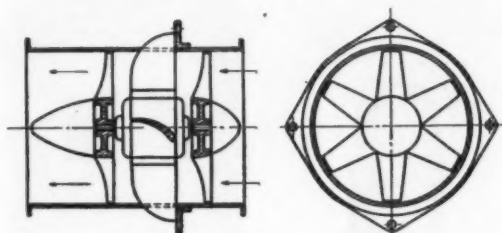


FIG. 3. EARLY ENGLISH FAN

increase the pressure developed and the fan efficiency. Such vanes can be arranged ahead of the wheel or after it, or both. One type is shown in Fig. 6.

#### CHARACTERISTICS

The typical characteristic curves of axial flow fans, the effect thereon of variations in several features of design, and their comparison with common types of centrifugal fans are shown in Figs. 7 to 10. In fairness to the centrifugal fan, it must not be overlooked that the axial flow fan derives many of its characteristics from close adherence to the principles of ideal flow and to maintenance of close manufacturing tolerances. The centrifugal fan with which the comparison is made is of the commercial type in common use where the factors given are often compromised to reduce cost. Were the centrifugal fan to be as precisely made as the axial flow, regardless of cost, the comparisons, especially of efficiency and noise, might easily be radically changed. Caution must be used in attempting to assign other than comparative or typical value to any such curves. With the many variations of design features, including a multiplicity of airfoil sections, variations in number, size, pitch and twist of blades, percentage hub and type and arrangement of inlet and diffuser vanes, the designer is presented with an infinite number of combinations from which to select. The result is that many characteristics desirable for an individual application may be built into an axial flow fan, although sometimes compromising other desirable characteristics. In addition, the boundaries of

the field of exploration of the effect of the various design features are wide, and positive limitations of characteristics resulting from untried combinations, difficult to define.

Fig. 7 shows the effect of various blade angles on the volume-pressure relationship with other design features held constant. The first *curve a* is with a flat pitch, and succeeding curves are with increasingly larger pitches. It will be noted that *curve a* shows a steadily rising pressure characteristic and that the other extreme *curve c* shows the pressure characteristic having a substantial dip. The resemblance to centrifugal fans is immediately apparent where the axial flow with a flat pitch has a characteristic comparable to a backward curve centrifugal fan. On the other extreme, *curve c* has a characteristic

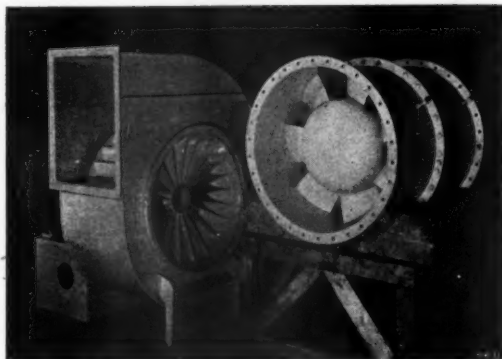


FIG. 4. COMPARISON OF SIZES

bearing a striking resemblance to a forward curve blade centrifugal fan. The unique characteristic of *curve c* is occasioned by the same circumstances in both the axial and radial flow fan in that wheel velocities are high and a majority of the pressure is made available by conversion of the high velocity pressures into static pressures. In the case of an axial flow fan, conversion is made in the diffuser vanes, and in the case of the forward curve fan, in the scroll volute. A fan having a characteristic such as *c* must be selected with caution to insure operation at a stable part of the curve. It would not be suitable for applications requiring a wide range of capacity at constant speed. On the other hand, and again similar to a forward curve centrifugal fan, it has the advantage of giving the most air volume at a given speed, or in other words, permits the use of a smaller unit at the same speed.

Fig. 8 shows the effect on the pressure volume relationship of the addition of inlet vanes and diffuser vanes. Vanes before the wheel or inlet vanes produce the effect of an increased speed and vanes after, commonly called diffuser vanes, raise the pressure curve and lower the specific speed. A propeller on an airplane, even when functioning at low angles of attack, would benefit somewhat from the use of similar vanes called contra vanes, if structural features did not prohibit their use.

Fig. 9 compares the static pressure, horsepower, and efficiency curves of a medium angle axial flow fan, a backward curve centrifugal fan, and a forward curve centrifugal fan, all selected to meet the same pressure-capacity conditions. Note that some similarity between the pressure curve of this design axial flow fan and a forward curve centrifugal fan still exists. A flatter pitch blade with higher speed or larger diameter would give a characteristic more nearly like the backward curve centrifugal.

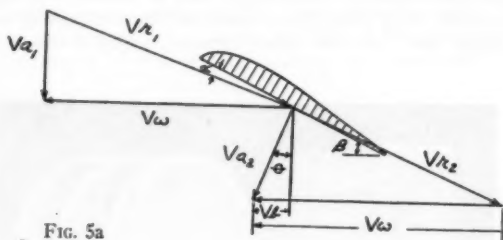


FIG. 5a

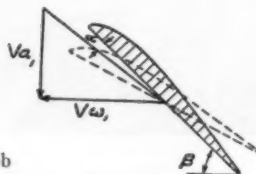


FIG. 5b

FIG. 5. BLADE VELOCITY VECTOR DIAGRAM

## Explanation of Fig. 5

The velocity vector diagram for any section through an axial flow fan blade is shown in Fig. 5a.  $V_{a1}$  represents the absolute air velocity into the wheel.  $V\omega$  is the tangential velocity of the blade which has a pitch angle of  $\beta$ .  $V_{r1}$  then represents the resultant or the air velocity relative to the blade.  $V_{r1}$  strikes the blade at angle  $\alpha$ , the angle of attack.

The air is shown passing through the wheel and leaves at relative velocity  $V_{r2}$  which combined with the tangential velocity  $V\omega$  gives an absolute air leaving velocity of  $V_{a2}$ . Note that this velocity is not axial in direction but deviates from the axial by an angle of  $\alpha$ .  $V_b$  thus represents the rotational component of  $V_{a2}$  which would be dissipated by the spiral motion unless recovered by guide vanes.

The need for a twisted blade is shown in Fig. 5b which takes a section nearer the center. Assuming the same absolute air velocity,  $V_{a1}$  will be the same, but the tangential velocity,  $V\omega_1$ , will be decreased in direct ratio to the radii. Thus angle  $\alpha$  would be decreased possibly to a negative value if the same pitch were used. An increased pitch as shown permits maintaining a suitable angle of attack. When angle becomes too great for optimum results, the blade is replaced with hub.

Likewise, for very small values of air flow,  $V_{a1}$  is decreased for the same value of  $V\omega$  and angle  $\alpha$  may exceed the angle of stall resulting in inefficient flow.

The horsepower curves are plotted for percentages of individual horsepower consumptions at the common pressure-capacity conditions. Consequently they indicate the variation in curve shape only, not relative power consumption at the rated condition. Note that the axial flow horsepower curve is approximately self-limiting, showing only a small rise at shut-off. Were a flatter pitch blade used, the self-limiting characteristics might be compromised.

The efficiency curves are drawn in per cent of maximum for each individual fan, and therefore show relative location of peak efficiencies only. The absolute values will vary with design but it can be generally stated that mechanical efficiencies of axial flow fans are comparable in value to those of centrifugal fans and may exceed them. However, since outlet velocities are frequently higher, the spread between static efficiency and mechanical efficiency might be greater. It should be especially noted that the peak efficiency of an axial flow fan occurs to the right of the curve, that is, toward free delivery. This fact automatically insures a stable operating point even with high pitch blades if the fan is selected to operate at best efficiency.

Noise was always a weakness of early forms of axial flow fans. High specific speed propeller fans could deliver pressure only by operating at exces-

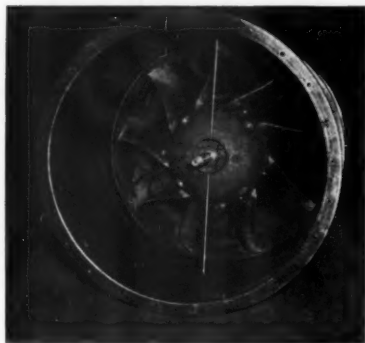


FIG. 6. INTERIOR OF AXIAL FLOW FAN

sive tip speeds and by operating to the left on the pressure-capacity curve. They were then out of their best sphere of use and either noisy or very noisy, depending on just how far afield they were.

Only after the introduction of medium pressure axial flow fans could moderate pressure ratings be met by fans operating within their proper sphere. Noise levels were more satisfactory, but only after extensive research were they reduced to a point comparable with centrifugal fans. As a result of this work, carefully designed medium pressure axial flow fans are today run at speeds which noise had always prohibited for exacting applications. Today within the medium pressure range encountered in ventilation, the axial flow fan may be as quiet as the centrifugal, and if carefully made, it may be even quieter. Such a fan demands mechanical perfection greater than now exists in a commercial centrifugal fan, so the cost of a refined axial flow fan often might not be justified by the use. As this perfection is sacrificed to price, the noise advantage of the axial flow fan over the centrifugal can quickly disappear. For this reason every axial flow fan, regardless of design cannot be indiscriminately considered quieter than a centrifugal fan. Tip speed is a poor

indicator of the respective noise merits of two different kinds of fans. A fan with one combination of design features may operate at a higher speed and yet have a lower noise level than a fan with other features. The only way to properly judge the sound ratings of different kinds of fans offered for a particular service is to compare their noise ratings. The noise produced by any given fan will vary greatly with the speed.<sup>1</sup> Since the reduction in speed means a reduction in pressure, it is imperative that those systems where noise

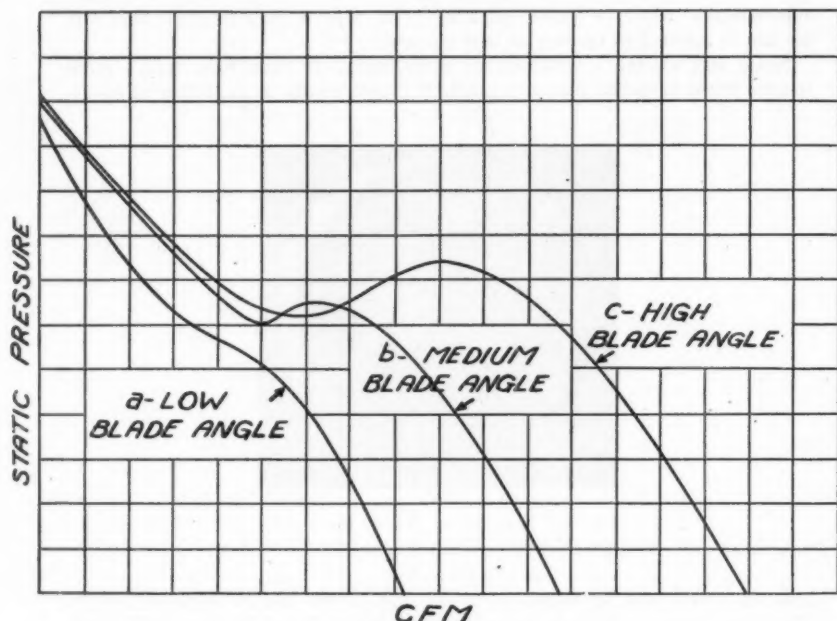


FIG. 7. EFFECT OF BLADE ANGLE

must be low should be designed for minimum resistances. Variable speed drives where feasible insure operating at all times at the minimum noise level consistent with the system demands. Maximum noise is thus generated only when maximum flow is required.

Noise consciousness will continue to increase so the noise level will become a more important factor in the economic selection of a fan. Noise ratings will put selection on a more tangible basis. System sound treatment is often economically warranted as with centrifugal fan installations, and the problems involved are similar, but the cost of treatment must be balanced against the

<sup>1</sup> For constant size, constant rating point.

$$\text{Db change} = 50 \log_{10} \frac{\text{RPM}_2}{\text{RPM}_1}$$

Thus, to double the speed of a fan will raise the noise level by 15 db.

difference in fan costs. Where sound treatment must be drastic enough to increase operating pressures, both installation and operating costs pyramid.

Comparative noise curves of axial flow and backward curve centrifugal fans are shown in Fig. 10. Superimposed are efficiency curves transferred from Fig. 9. Note that the low noise level on the axial flow fan is to the right of the curve and that there is a sharp rise of approximately 10 db near the center. This is characteristic of both low and high angle fans. The backward curve centrifugal fan has a broader sound range devoid of critical points. For both

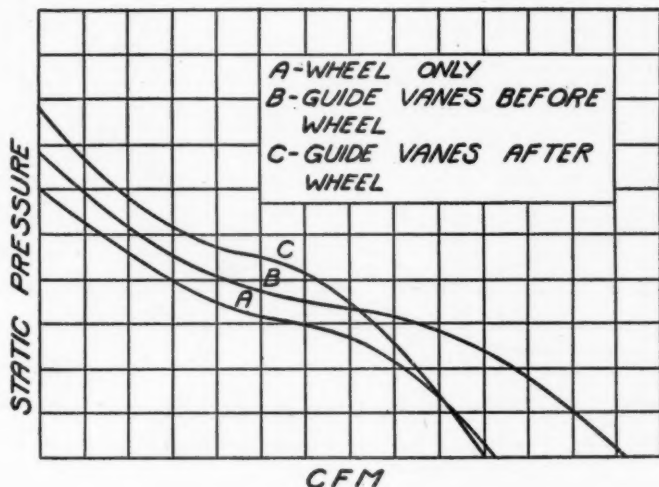


FIG. 8. EFFECT OF GUIDE VANES

types the region of minimum noise level coincides with the region of maximum efficiency.

All these characteristic pressure, horsepower, efficiency and noise curves evidence that the optimum operating point of an axial flow fan is nearer free delivery than of the centrifugal and in that region even a high pitch fan gives stable operation. However, if the system characteristic of the application is varying or indefinite, a centrifugal fan insures more trouble-free operation.

#### APPLICATION

In applying axial flow fans to systems requiring little or no static pressure, performances better than with centrifugal fans can be obtained. However, a propeller or disk fan is usually indicated, since its lower cost more than offsets any loss in efficiency.

Maximum pressures for which axial flow fan may be designed cannot be categorically stated, especially if multirotor units are included. Limits are, however, well above the ventilation range and therefore beyond the scope of

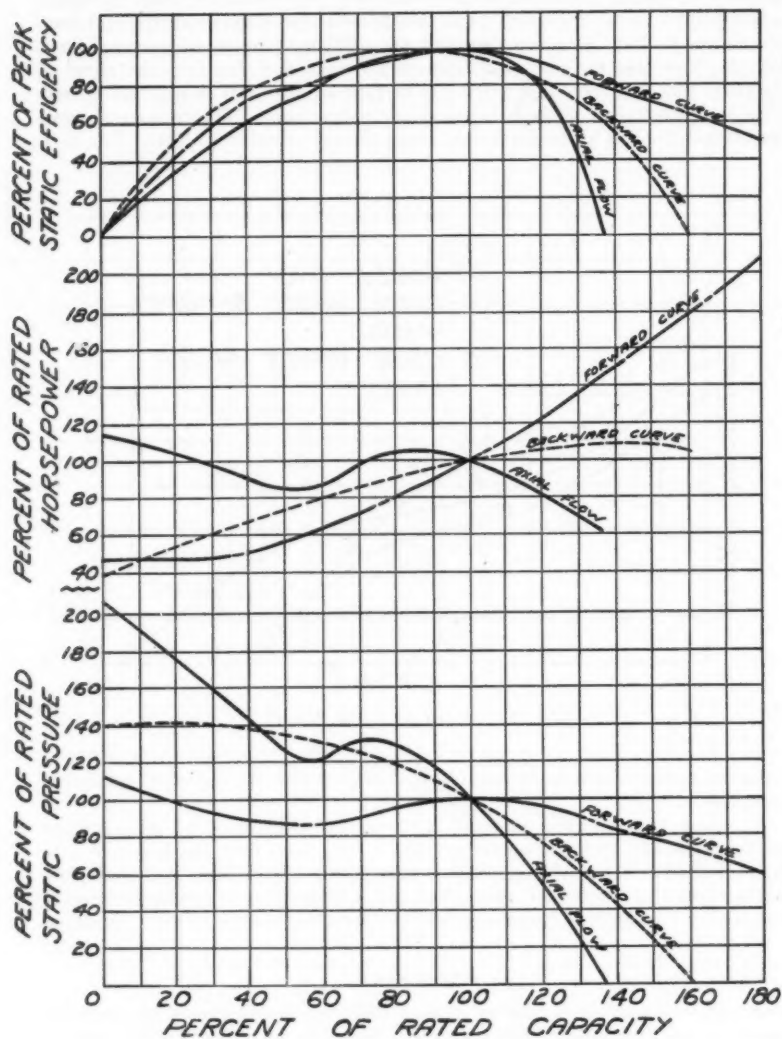


FIG. 9. COMPARISON OF CHARACTERISTIC CURVES

this paper. Suffice it to say that a single stage fan of the general construction under discussion is capable of meeting any pressure demand imposed by any ordinary ventilation requirement.

Pressure alone, however, is not a true criterion of the proper fan selection. The pressure-capacity-speed relationship must be considered. In order to see how the axial flow fan fits into the fan family as it now exists, it is necessary to consider specific speed.<sup>2</sup> Specific speed is a non-dimensional indication of

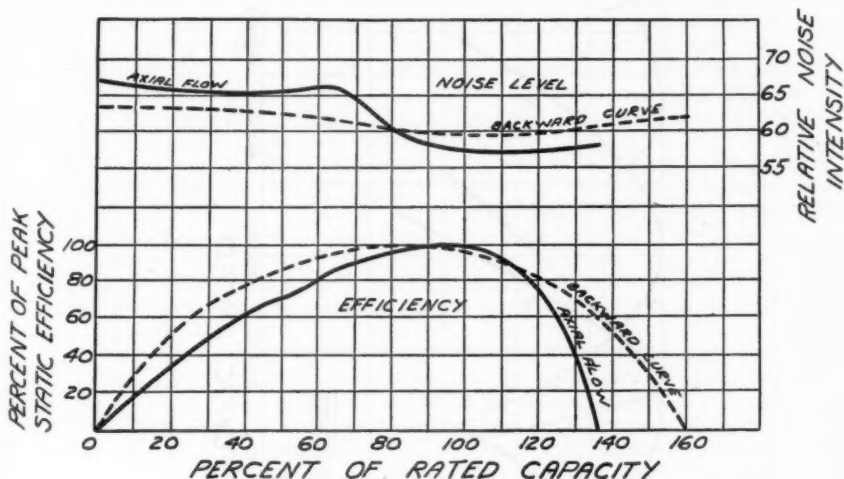


FIG. 10. COMPARISON OF NOISE CURVES

such pressure-capacity-speed relationship and may be defined as the revolutions per minute at which a fan of any one type would operate to furnish 1 cu ft of air per minute at 1 in. pressure. Numerically, this would be:

$$N_s = \frac{\text{RPM} \sqrt{\text{CFM}}}{(P_s)^{3/4}}$$

where

$N_s$  = specific speed.

RPM = rev. per min. of fan.

CFM = cubic feet per minute of fan.

$P_s$  = static pressure of fan in inches of water.

From the formula, values of specific speed may be computed for each point on the pressure-capacity curve. Disregarding slight changes in efficiency with fan size, these values would then hold for any one design of fan running at any speed regardless of size. These values may then be plotted against any other characteristic such as efficiency whence from the pressure-capacity-speed requirements of any application, the efficiency of the correct size of any type of fan may be determined. When such curves are drawn for a complete family

<sup>2</sup> For a more complete discussion of specific speed see *Fan Engineering*.

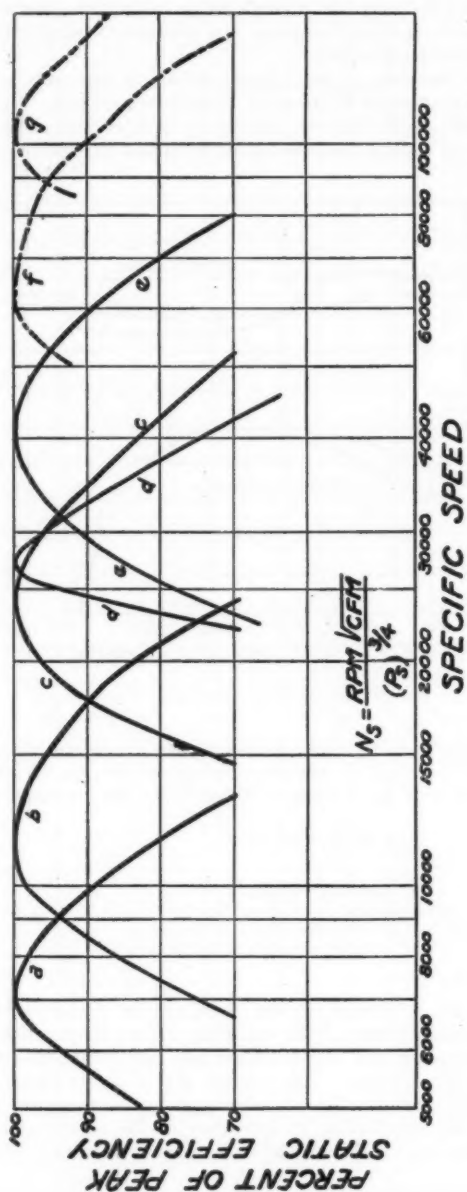


FIG. 11. SPECIFIC SPEED—EFFICIENCY CURVES

of fans, the best fan selection for any given condition may be readily determined. Such a group curve is shown in Fig. 11.

This curve is plotted between per cent of individual peak static efficiencies and specific speeds for several types of centrifugal and axial flow fans. *Curve a* indicates a very high pressure-low capacity blower such as a turbo compressor or supercharger. *Curve b* represents a high pressure blower, *curve c* a medium pressure fan commonly used for mechanical draft, *curve d* a forward curve ventilating fan, and *curve e* a backward curve ventilating fan. It is thus seen that the range of specific speed from 0 to 60,000 is well covered by centrifugal fans operating at optimum efficiencies. If axial flow fans were not available, 60,000 specific speed would be about the limit of good efficiency. If the pressure-capacity-speed requirements of an application called for higher than 60,000, it would be necessary to drop the speed and use a larger fan to fall within an efficient operating range.

Also plotted are two representative axial flow fan curves *f* and *g* showing their normal range of specific speed to be above 60,000. With this range covered by efficient fans, it is now frequently possible to increase the speed of an application requirement and use a smaller fan at less cost. The conclusion should not be drawn from these curves that axial flow fans cannot be designed for specific speeds below 60,000, but rather that above that limit they are definitely indicated. Conversely, at low specific speeds the centrifugal fan is definitely indicated.<sup>3</sup>

Obviously, there is an overlapping region where considerations other than efficiency decide the choice. Such considerations are mainly cost, space, and reliability. Future costs are difficult to predict, but at present there is evidence that when both types of fans are made comparable in performance, they compare in cost. In this respect it must be remembered that while the axial flow fan is smaller, lighter, and frequently runs at higher speeds, it must be more precisely made, and this precision is reflected in the manufacturing cost. The smaller space requirements of an axial flow fan will frequently be the deciding factor since space saving is the axial fan's chief virtue. That feature alone dictates its selection in many cases where other factors might otherwise indicate the choice of a centrifugal fan. As for reliability of service, installations handling contaminated air or where maintenance shutdowns disrupt vital service are better left to the centrifugal fan. Furthermore, the centrifugal fan is better adapted for applications where hot gases are handled. Axial flow fans are used to handle heated air but the factors in favor of their use must bear an additional burden to offset some sacrifice in reliability and serviceability and an increase in cost. As for reliability of performance, the pressure, horse-

<sup>3</sup>Example:

Given: Fan to handle 3,000 cfm at 3 in. static pressure, direct connected to 1750 rpm motor.

$$\text{Compute: } N_s = \frac{1750 \sqrt{3000}}{(3)^{3/4}} = 42,000$$

42,000 specific speed falls within a range suitable for a centrifugal fan which would then be the obvious selection although the use of an axial flow might not be entirely excluded. If space saving were important, the obvious attempt should be made to decrease the size by increasing the speed. Such a change might also result in a lower cost. Increasing speed to the next higher motor speed of 3500 rpm—

$$N_s = \frac{3500 \sqrt{3000}}{(3)^{3/4}} = 84,000$$

84,000 specific speed would be outside the range of good efficiency for a centrifugal fan and an axial flow fan would be definitely indicated.

With the data given, investigation should then be made of the relative noise levels, space requirement, power consumption and cost, to determine the best selection.

power and noise characteristics of a backward curve centrifugal fan are optimum where the system resistance is indefinite or variable

One prime application where space conservation is requisite is in mine ventilation, since air quantities are often extremely high and pressures comparatively low. Fig. 12 shows an early mine fan installed above ground, but underground installations present the greatest opportunity for economy. The saving in cost of excavation alone may easily decide the choice. A comparable application might be in future vehicular tunnels where specific speeds are suitable for axial flow fans and expensive buildings must be constructed just to house the ventilating equipment.

As is only fitting, the aeronautical laboratories and airplane engine plants have found many uses for the axial flow fan. See Fig. 13. A perfect example

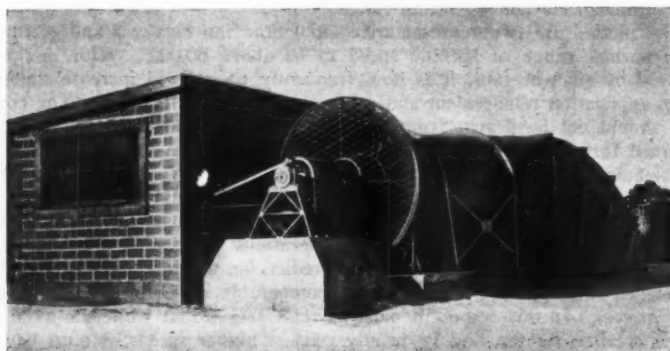


FIG. 12. MINE FAN INSTALLATION

is for drawing cooling air over engines undergoing performance or run-in tests. With the engine at one end of the engine mount and an axial flow fan at the other, pressure losses and installation costs are minimized. A similar cooling use is for large vehicles such as diesel locomotives and tanks where the amount of work to be done and the space value preclude the use of any less efficient or more bulky type of fan.

Of course, the ideal application for axial flow fans is on ships, where their small size, light weight, high efficiency and quiet operation are so advantageous. By using high speed motors, units can be built to fit directly in main trunks with little change in duct design. Navy requirements and the foresight of naval engineers provided the real impetus for axial flow fan development on a large scale. Quiet operation with accompanying improvement in efficiency, decrease in weight, and broadening of pressure ranges was acquired without sacrificing ruggedness, since naval installations must withstand concussion and shock. Details of this application are given in a recent paper.<sup>4</sup> With the vast quantity of fans required, completely standardized units could be adopted, but only after the naval engineers felt the optimum in performance had been attained. As a result of this insistence on excellence before standardization,

<sup>4</sup> Warship Ventilating, Heating and Air Conditioning, by Comdr. T. H. Urdahl and W. C. Whittlesey. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 35.)

axial flow fans will be available in many sizes and designs for post-war industrial applications and their successful operation will have been proved.

In the future there will be many ventilating applications where the axial flow fan will replace the centrifugal largely on the basis of space saving. The air conditioning industry will find many places for this fan, as will the drying and electrical industries. In fact, nearly every industry will find it worthy of consideration.

In conclusion, it might be said that the true axial flow fan was born in the aeronautical research laboratory, received its basic training in the Navy, and is ready to join the industrial army. It has fields of application where its use

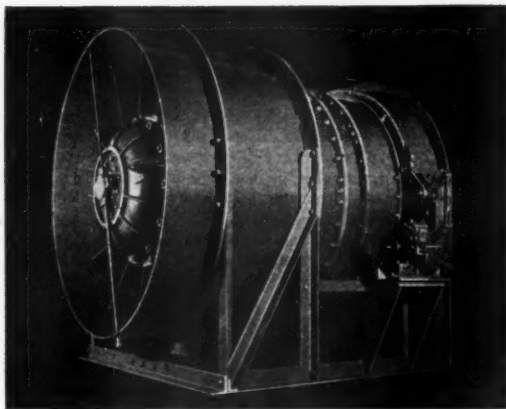


FIG. 13. ENGINE VENTILATING FAN

is definitely indicated but it can also be misapplied. It fills a gap in the specific speed chart by providing a high efficiency unit between centrifugal fans and propeller fans. Its range overlaps both of these sufficiently so that it will replace many disk and propeller fans and some centrifugal fans. It is not, however, a universal fan to be indiscriminately applied, and the choice between an axial flow and a centrifugal fan must still be intelligently made by the engineer.

## DISCUSSION

H. E. ZIEL, Detroit, Mich.: I would like to inquire as to the application of the axial flow fan for engine cooling in a test cell. It was not clear to me from the slide whether the fan unit was drawing or discharging air over the engine.

At the present time engine cooling is usually accomplished in a test cell with a multivane fan unit located some distance from the engine, the fan unit discharging the air through a 48-in. duct. This duct terminates near the hub of the propeller, the air being discharged through the propeller over the engine. This method of cooling has proved satisfactory except for the interference caused by the concentric duct in the cell which interferes with the monorail system for handling the engine. My thought was that if this axial flow fan unit could be placed directly in front of

the propeller and blow through the propeller for engine cooling, it would eliminate the ductwork required in the cell when using a multivane fan unit.

**AUTHOR'S CLOSURE:** We cannot say too much as to just how any one system is arranged, but in this particular case the air was being exhausted from the engine. Air went through the engine first, and then to the fan. This fan could also be arranged to blow through the engine. In such a case it would be best not to have the fan too close to the engine, for it might influence the air flow into the propeller, and therefore it might affect the test results. At least a short duct, as used with the centrifugal fans would give better results.

When the fan is placed on the back end, on the exhaust side of the engine, the fan motor is the critical feature. With hot air passing over it, you have to pay proper attention to the design of the motor.

Speaking generally, the axial flow fan can be adapted to either exhausting or blowing arrangements for engine testing.

Since presentation of this paper the *National Association of Fan Manufacturers* have adopted nomenclature for various types of axial flow fans:

Propeller Fan consists of a propeller of disc type wheel within a mounting plate or ring and including driving mechanism supports either for belt drive or direct connection.

Tubeaxial Fan consists of a propeller or disc type wheel within a cylinder and including driving mechanism supports either for belt driver or direct connection.

Vaneaxial Fan consists of a disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports either for belt drive or direct connection.



**1251**

## THE AERODYNAMIC DEVELOPMENT OF AXIAL FLOW FANS

By T. H. TROLLER,\* AKRON, OHIO

### DEVELOPMENT UP TO THE PRESENT

THE AXIAL flow fan, as it is known at present, consists of one or several rotating wheels, carrying twisted, airfoil shaped blades on a round hub, and of counterrotating means suppressing the twist in the air set up by the rotating wheel. The elimination of air rotation behind the fan wheel permits building up of high pressure by a fan conveying air essentially straight through a duct.

The basic form of the axial flow fan has been well known for about fifteen years. The aerodynamic design of axial flow fans is an achievement of aerodynamic science. It is founded on the employment of airfoil sections, the characteristics of which have been well established in a great number of wind tunnel tests conducted in the course of aeronautical development during the last 35 years. These characteristics of airfoils, available without specific tests for the use of fan design, were the only experimental data needed. From them rational analysis produced the aerodynamic design of the axial flow fan. The analysis started with the recognition, which in the author's belief was first pointed out in the Russian literature,<sup>1</sup> that in an axial flow fan a continuous flow of *bound circulation* tied to the blades, hub, vanes and the outside wall must exist in such a way as to form a continuous flux corresponding to certain magnetic circuits in electric machinery.

*Circulation* in aerodynamics is defined as the line integral

$$\oint v \cos \theta \, ds$$

along a line enclosing a space in a moving fluid and *bound circulation* is this integral value when taken in the fluid around a solid core to which this circulation appears attached.

In the formula:

$v$ —Velocity at any point of the closed line.

$\theta$ —Angle between velocity and this line of element.

$ds$ —Element of the line.

\* Director of Research, Daniel Guggenheim Airship Institute. Consultant, La-Del Conveyor & Mfg. Co. Member of A.S.H.V.E.

<sup>1</sup> *Theory turbilonnaire de l'hélice propulsive*, by N. E. Joukovsky. (Original in Soc. Math. Moscou, 1912, Reprint Paris 1929.)

Presented at the 50th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1944.

Various publications<sup>2,3</sup> appearing about 1930 connected this thought with the investigation of rotating airfoil wheels in fans until in other publications complete working schedules were developed for the design of the fan structure.<sup>3,4</sup> The very first fans designed according to this scheme proved entirely satisfactory as far as efficiency, stability of the characteristics, simplicity and size of the resulting structure were concerned.

The further improvement and refinement of the theory which appears desirable and to which much thought has been given in recent years has for various

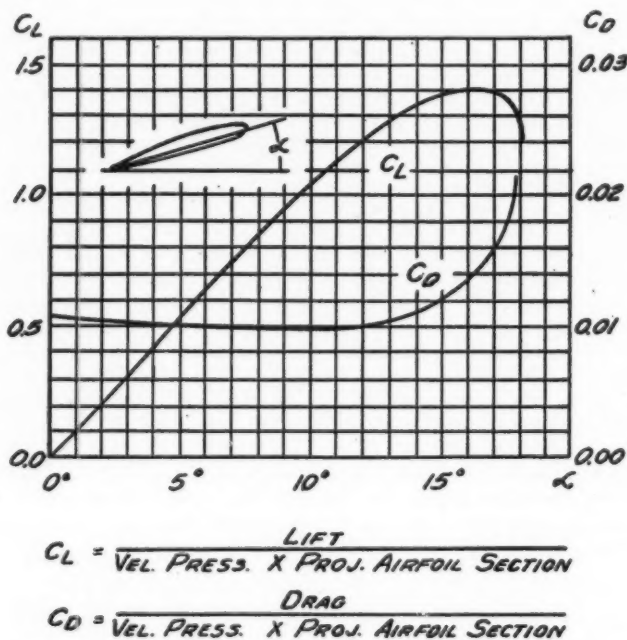


FIG. 1. CHARACTERISTICS OF AN AIRFOIL SECTION SUITABLE FOR FAN BLADE DESIGN

reasons not yet resulted in a further technical advancement of the axial flow fan. Also, specific investigation of airfoils and airfoil grids for the purpose of fan design have so far not opened any new ways for improving fans within the range in which they now are widely used.

However, an approach to the problem in which the aerodynamic computation of the forces on the blades is replaced by design specifications based on

<sup>2</sup> Diagrams for Calculation of Airfoil Lattices, by Albert Betz. (NACA Tech. Mem. No. 1022, 1931.)

<sup>3</sup> Axialgebläse vom Standpunkt der Tragflügeltheorie, by Curt Keller, 1934. (Gebr. Leeman & Co., Zurich, Switzerland.) Partial Translation—Axial Flow Fans, Keller-Marks, McGraw-Hill Book Co.

<sup>4</sup> Zur Berechnung der Schraubenventilatoren, by T. H. Troller. (On The Design of Axial Flow Fans—1931.) Abhandl. aus dem Aerod. Inst. Aachen, Heft 10—Verlag Springer, Berlin, 1931.

the Eulerian conditions for the flow in rotating channels has been advanced recently and constructions based on the ensuing design rules are successful for blowers in the extremely high pressure range.<sup>†</sup>

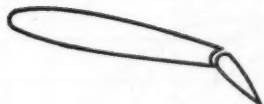
The one often heard objection to the axial flow fan is that it is noisy. There were and still are axial flow fan installations in existence which are unsatisfactory in this respect. The causes of high noise in these cases may be: over-



#### SLOTTED AIRFOIL

$C_L$  = VALUES UP TO 2.0

$C_D$  = MODERATELY INCREASED



#### AIRFOIL WITH FLAP

$C_L$  = VALUES UP TO 2.5

$C_D$  = STRONGLY INCREASED



#### HOLLOW AIRFOIL WITH BOUNDARY LAYER REMOVED BY SUCTION

$C_L$  = VALUES UP TO 5.0

$C_D$  = LOW, BUT SPEC. ENERGY FOR SUCTION IS NECESSARY

FIG. 2. VARIATIONS FROM PLAIN AIRFOIL SECTIONS

emphasis on fan compactness and therefore unduly high fan speed; very turbulent air inflow to the fan; aerodynamically or mechanically improper fan design; and, in the minority, inherent conditions of the fan operation. As far as such inherent features of the axial flow fan are concerned, however, there is nothing discernible to handicap the average axial flow fan in comparison with older types of air conveyors of comparable size.

#### EXPECTED FUTURE PERFORMANCE

A look into the future of fan development suggests investigation of certain absolute limits of performance such as: maximum pressure obtainable in one stage, maximum pressure for a given diameter, maximum pressure for a given speed of rotation, maximum pressure obtainable in a fan of given length, maximum efficiency, minimum noise for a given volume or pressure, etc. An

<sup>†</sup> Wattendorf, F. L., The Ideal Performance of Curved Lattice Fans, by Karman. (Anniversary Volume, California Institute of Technology, 1941.)

investigation of the limits of the fan in this manner, however, is academic since there is no practical application in which the final choice of the fan design does not constitute a compromise between various aspects such as size, cost of construction, cost of operation and noise level. However, an analysis of a few of these limits as isolated features is of interest.

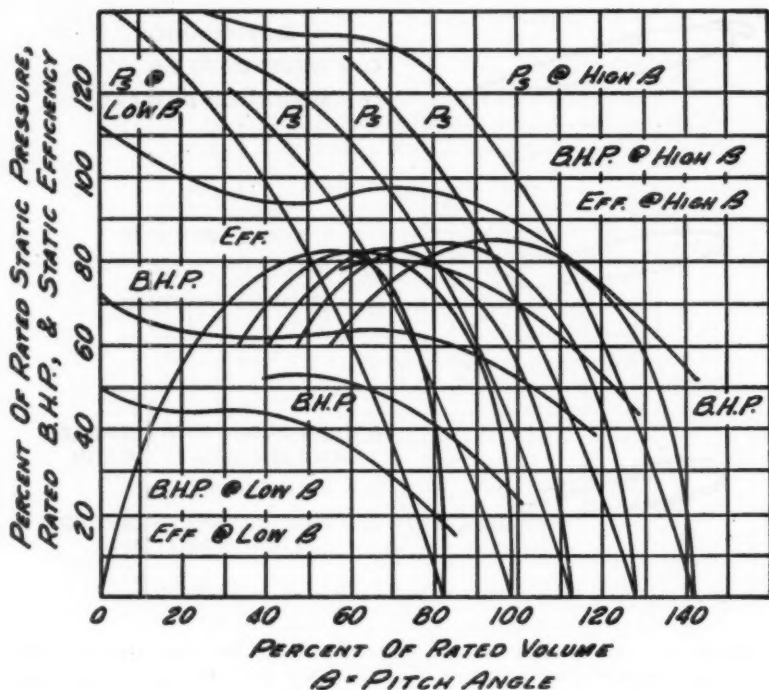


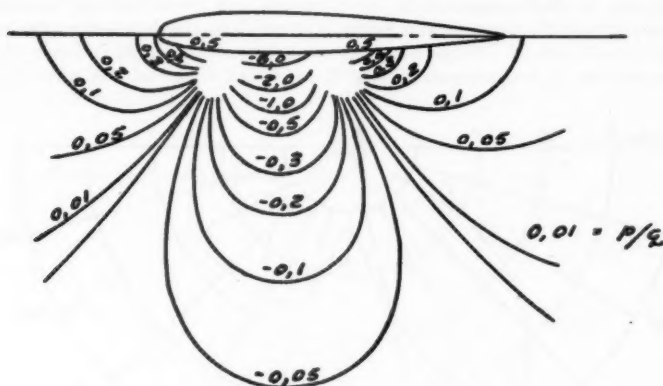
FIG. 3. CHARACTERISTICS OF AN ADJUSTABLE PITCH AXIAL FLOW FAN

For an understanding of the action of the axial flow fan it is necessary to consider the basic characteristic of a curved surface or airfoil section when moving through air. Such a typical characteristic of an airfoil, representing the forces on the airfoil as a function of the angle of attack, is shown in Fig. 1. Two aspects in particular are noteworthy:

1. There is an almost linear increase in the lift force, which is the force perpendicular to the relative motion between surface and air, in existence over a range of angles of approximately 16 deg. Beyond this range the lift function deviates from the straight line, then a peak in the lift force is reached, and, finally, a reversal or stalling of the lift line occurs.

2. The drag of the airfoil or force opposite to the direction of relative motion is small relative to the lift force obtainable over the major range of the straight relation range between angle and lift force.

The linear connection between angularity and lift force re-appears in the fan characteristics as an almost linear connection between volume and pres-



$p$  = STATIC PRESSURE  
 $q$  = VELOCITY PRESSURE

FIG. 4. LINES OF CONSTANT PRESSURE ABOUT A SYMMETRIC AIRFOIL AT ZERO ANGLE OF ATTACK

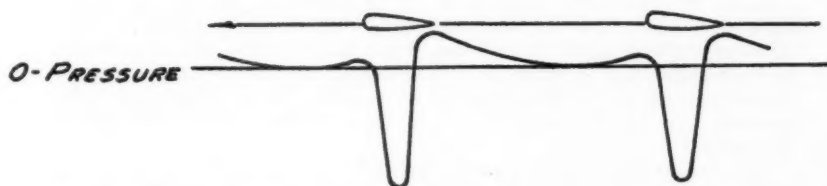


FIG. 5. PRESSURE FIELD ALONG A LINE NEAR AN AIRFOIL SECTION PARALLEL TO ITS PATH OF ROTATION

sure in the fan over a wide range. The peak re-appears in the fan characteristic as a stalling point or at least as a definite bend in the characteristic pressure-volume curve of the fan. The exact nature of the effect of the *stalling* of the airfoil sections forming the fan blades depends largely on the angle setting of the blades against their plane of rotation and also on some other factors. The low drag coefficient of the airfoil sections means essentially that in the angle of attack range of the blade sections, where these

favorable conditions exist, small losses and therefore high efficiencies are obtainable in the axial flow fan.

This characteristic of the airfoil taken from wind tunnel tests determines maximum pressure and maximum efficiency that can be obtained in a fan since it is little different from the characteristic of the blade sections employed in fan blades. The maximum lift coefficient of the fan blades is essentially comparable to that measured in wind tunnels and, therefore, the maximum pressure obtainable from a fan blade is limited to the lift force obtainable on an airfoil section. There are tests available which would indicate higher lift peaks on airfoil sections arranged in a fan than on the same airfoil

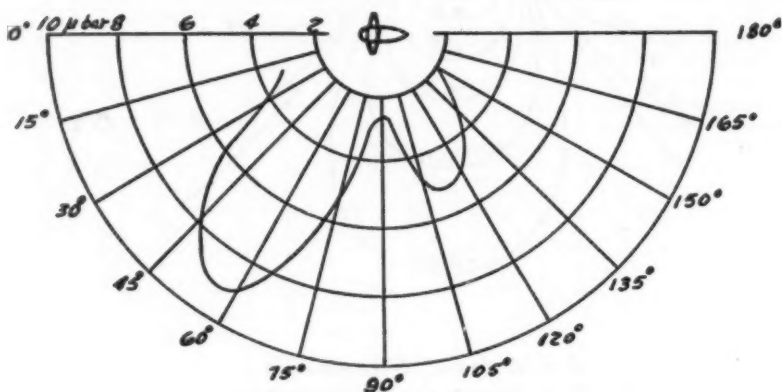


FIG. 6. SOUND PRESSURE FIELD ABOUT A TWO BLADED PROPELLER (LINE INDICATING PRESSURE ALONG A CONCENTRIC CIRCLE ABOUT THE PROPELLER)

measured by itself in the wind tunnel. These differences, according to present knowledge, would, however, not have a large effect on the behavior of a fan. As an approximate figure we may state that a fan wheel can develop a pressure approximately equal to 1.2 to 1.6 times the dynamic pressure of the rotation-velocity existing at the hub section of the propeller wheel. Somewhat higher pressures can be obtained with the narrow channel wheels described by Wattendorf.

The absolute limitation in pressure obtainable in one wheel is given by that blade velocity which forms the limit for efficient action of airfoils with lift forces corresponding to this pressure peak of 1.2 to 1.6 times the velocity pressure on the blade. This limiting velocity, depending somewhat on the shape of the section is equal to approximately 80 per cent of the velocity of sound or about 880 fps. A fan wheel running with such a tip velocity could develop roughly 135 in. water pressure, assuming the hub diameter to be equal to 0.9 times the tip diameter of the blades.

Several means are known for increasing the peak lift developed on an airfoil beyond the value shown in Fig. 1. They consist of additions to or variations from a plain airfoil such as shown in Fig. 2. The highest lift peaks are obtainable with airfoil sections using *boundary layer suction* which

would bring about peak lifts up to three times that available with a plain airfoil or curved section. Any employment of such means will of necessity complicate the otherwise striking simple appearance of the axial flow fan. The *boundary layer suction* would consist in the removal of a thin layer of air, the so-called boundary layer, from the surface of the blade, by taking it through openings into the inside of the blade and then discarding it again at a suitable location. It should be expected that developments of this nature will appear in the future and that with such developments increased pressures in one wheel will be used for a given turning speed of a fan. Regarding the velocity limit, at which such a special blade section can be successfully applied, it should be observed that the very high  $C_L$  values with simultaneous

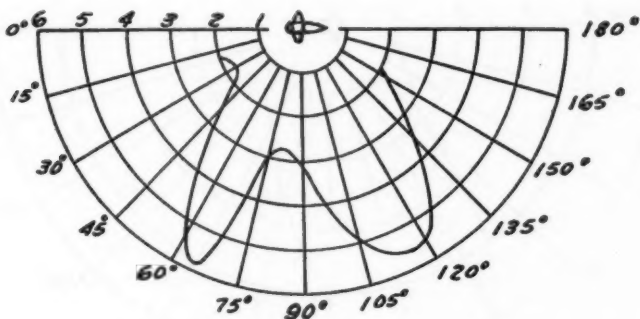


FIG. 7. SOUND PRESSURE FIELD ABOUT A THREE BLADED PROPELLER  
(SAME AS IN FIG. 6)

low  $C_D$  values would be possible only for a range considerably lower than 0.8 times the sound velocity.

The efficiency limit as it would appear in the light of the airfoil theory would be represented by the formula for one blade section at a radial station. (For the fan as a whole the efficiency would be given by integration over the whole radius. In the simple form given here an average section is substituted for the whole blade.)

$$\eta = \frac{1}{1 + \epsilon/\lambda}$$

where

$\epsilon$  = ratio of drag to lift for the average blade section.

$\lambda$  = ratio of the velocity component of the air relative to the fan blade in direction of the fan axis to the velocity component in peripheral direction of the relative movement.

$\eta$  = fan wheel efficiency.

Assuming that energy losses in the fan outside of those due to the friction on the blades are relatively small, this efficiency formula would indicate that a peak of efficiency is reached when  $\lambda$  is approximately equal to 1. This would mean the airfoils stand at nearly 45 deg against the plane of rotation. Introducing for  $\epsilon$  a value of 0.01 as measured in wind tunnels, possible

efficiencies of about 98 per cent for the fan wheel itself are obtained. If to the losses occurring in the wheel are added those losses which have to occur in the countervanes or other counter-rotating means and in the rest of the motion of the air through the fan, it can still be assumed that ideal efficiencies of 95 per cent would be feasible if obtaining such efficiencies were the only consideration in the design of a fan.

As far as driving the efficiency of a fan to the theoretical limitation is concerned, it can be stated that a rapid increase in bulkiness and cost of the fan structure occurs beyond a point which lies between 75 per cent to 85 per cent fan efficiency to such an extent as to force in most cases the limitation of the efficiency to this range or, at any rate, to a value of under 90 per cent.

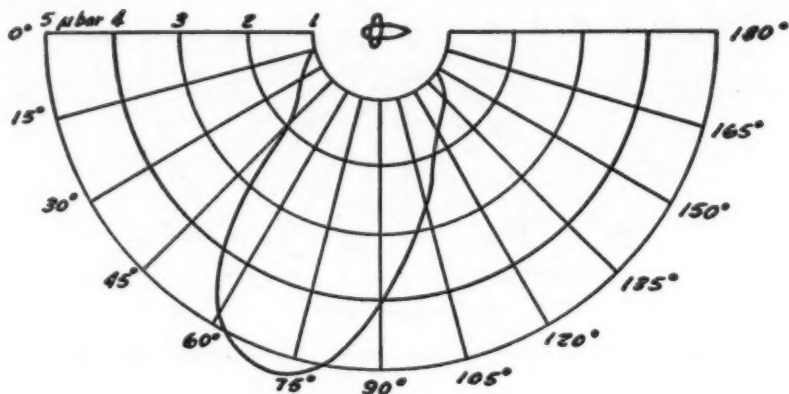


FIG. 8. SOUND PRESSURE FIELD ABOUT A FIVE BLADED PROPELLER (SAME AS IN FIG. 6)

Before leaving the discussion of fan pressure and efficiency, there is one other feature of the axial flow fan that merits mention regarding future development, namely, the possible increase of flexibility and of operating range of an axial flow fan by means of pitch adjustment of the blades. Pitch adjustment has become common for propellers of most airplanes and we may expect that a corresponding development will bring the advantages of adjustable pitch to many axial flow fan applications.

Fig. 3 shows pressure-volume characteristics, power-volume characteristics, and efficiency-volume characteristics for an axial flow fan with adjustable pitch at a number of pitch settings for a fan of 36 in. diameter delivery at rated capacity of 18,500 cfm against 4.5 in. static pressure. Any one single curve relating the power to the air delivery at a set blade pitch angle becomes the flat power curve generally existing at fixed pitch. This flat power curve of the fixed pitch axial flow fan permits the designer to dimension the driving motor essentially for the power required at the main operating point. However, there is no immediate saving in power consumption at low deliveries of air when such a change from the main operating point of the fan character-

istic is required at some period in the life of the fan. To obtain a saving in power or to change air delivery through a given duct system in a desired manner without regulation involving losses, pitch setting of the fan blades is employed most successfully, as indicated by the resulting characteristics of Fig. 3.

It is the author's opinion that an appreciable range of regulation for axial flow fans can be accomplished only by means of pitch setting of the blades and further, that the extension of the operating range by means of setting of the stationary vanes is restricted to such small limits that it appears to be of doubtful advantage. The pitch setting of the rotating blades is, of course, more involved mechanically than that of the stationary vanes but whenever

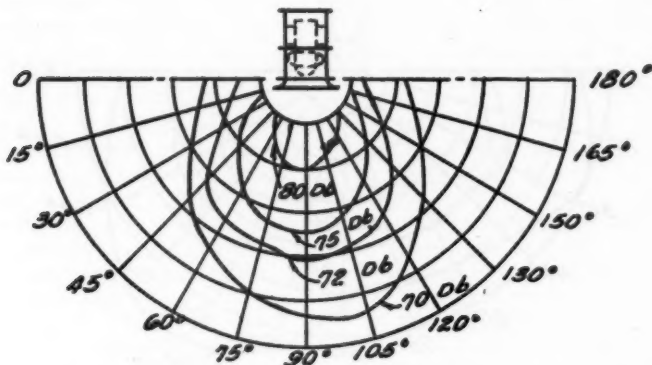


FIG. 9. SOUND LEVEL FIELD ABOUT A TWO-BLADED AXIAL FLOW FAN REVOLVING AT 3500 RPM (EQUAL NOISE LEVEL LINES AS FUNCTION OF THE LOCATION RELATIVE TO A FAN EXHAUSTING FROM A PLENUM)

an additional complication of the fan is worthwhile, it should also justify a mechanism for setting the pitch of the blades. This pitch setting of the blades can be accomplished by various designs and it can be applied: by setting the blades once during the fan installation to fit them to ducts; or by setting the pitch with simple mechanical means in rest periods of the fan; or finally by setting the pitch on the running propeller by hand or automatic steering devices. The type of mechanism adjustment to be chosen in each case would depend upon the purpose of the individual fan installation.

#### AERODYNAMICS OF FAN NOISE

In addition to forming the basis of the pressure and efficiency computations, the aerodynamics of the airfoil also explain the necessary appearance of noise in axial flow fans. If contributions to the noise due to imperfections in the fan structure are neglected, the fact remains that a source of sound is inherently connected with the rotating fan blades. Each airfoil is surrounded in its movement through the air by a field consisting of regions of lowered and raised air pressures (see Fig. 4 for the simple case of a symmetric airfoil

at zero angle of attack). An airfoil or a series of airfoils arranged in a rotating disc brings about a field of periodically changing pressures in any one point of the space in the immediate surroundings of the propeller disc (see Fig. 5). This periodically changing pressure field constitutes a source of sound waves which will emanate from the space of pressure disturbances. The sound-pressure field due to this peculiar source of sound is of more complicated form than, for instance, that of a simple membrane. At any reference point at a distance where the immediate pressure changes due to the moving blades could no longer be recorded sound waves arrive from the regions of periodically changing static air pressures and densities. These sound waves interact with each other so as to form a noise-level pattern based on the

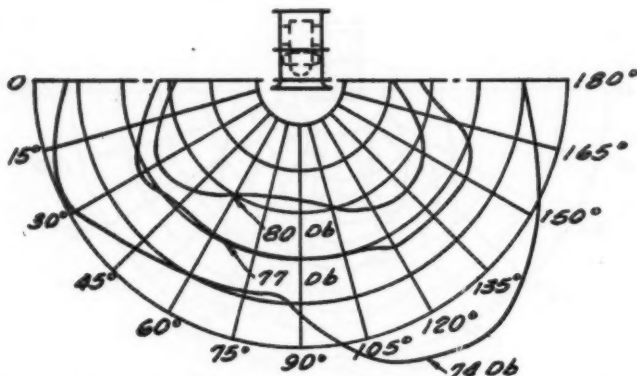


FIG. 10. SOUND LEVEL FIELD ABOUT A FOUR-BLADED AXIAL FLOW FAN REVOLVING AT 3500 RPM (EQUAL NOISE LEVEL LINES AS IN FIG. 9, FAN OF FIG. 10 IS NOT EQUAL IN SIZE AND PERFORMANCE TO THAT OF FIG. 9)

frequency of the emanated sound and on the shape of the pressure field surrounding the fan wheel.

The main frequency of the necessary noise is equal to turning speed times the number of blades. The shape of the pressure field depends on the number of blades and on a radial distribution of the pressure on the blades as well as on the peripheral pressure distribution along the path of the rotating blade sections and, further, also on the fan structure surrounding the rotating wheel.

The resultant sound level pattern contains lines which are radial at large distance from the fan and somewhat curved in its close vicinity, along which the measured sound reaches considerably higher values than at points between their angular sections at equal distances away from the fan center. The existence of such sound level distribution patterns is known from theoretical and experimental investigations of airplane propellers.<sup>5, 6, 7, 8</sup> Typical distributions

<sup>5</sup> The Source of Propeller Noise, by W. Ernsthäuser. (NACA Tech. Mem. No. 825.)

<sup>6</sup> Neuere Untersuchungen über das Luftschraubengeräusch, by W. Willms and W. Ernsthäuser. (Luftwissen 1938, pp. 128-134.)

<sup>7</sup> Investigations of the Origin of the Sound Emitted by Revolving Airscrews, by Obata-Yosida. (Rep. of the Aero Res. Inst. Tokyo Un. No. 132.)

<sup>8</sup> On the Direction of Properties of Propeller Sound, by Obata-Yosida. (Rep. of the Aero Res. Inst. Tokyo Un. No. 134.)

for two-, three-, and five-bladed propellers are shown in Figs. 6, 7, and 8 in which lines of equal sound pressure are given as recorded in airplane propeller tests. In these tests noise peaks for a two-bladed propeller, occurred under an angle of 55 deg, and then at 110 deg, measured from the forward extended axis of the rotating wheel in a plane through the propeller shaft. For a three-bladed propeller of the same speed and diameter and similar blade design, the peaks were obtained at 65 deg and 115 deg. For a corresponding five-bladed propeller the peak was obtained at 75 deg. The indicated noise levels are those of the main frequency of noise. Contributions of frequencies other than the main source of noise are in existence and would have a pattern of their own. Such other tones are partly related to the turning speed of the fan and are partly independent of the fan speed. In the latter class we have in particular the oscillations occurring in the frictional wake behind each rotating blade as a necessary source of noise. These other components are, however, usually of minor importance.

Compared to that of an airplane propeller the sound level pattern around an axial flow fan exhausting from or blowing into a plenum chamber is further complicated by the fan housing and particularly by a bell shaped intake. Some preliminary checks regarding such noise patterns for fans exhausting from a plenum are shown in Figs. 9 and 10. They bring out essentially the same character of the sound field as obtained theoretically and experimentally for airplane propellers even though these particular tests suffered from somewhat unsatisfactory test conditions in the laboratory. The existence of such a sound pattern should be taken into consideration in the establishing of testing specifications for fans because it may well be possible by choice of the number of blades to avoid pressure peaks at arbitrary points specified for such sound measurements without improving the noise characteristic of the fan throughout the space.

The noise level produced by a fan of given geometric outline is closely proportional to the tip speed of the blades, except for those noise components not inherently connected with the fan action.<sup>9</sup> Measured in decibels it is, over a wide range, a linear function of the logarithm of the tip speed, with the proportionality factor depending on distance from the fan, fan size, etc. The absolute lower limit of the noise level of a fan would perhaps be given by that size and velocity of the blades at which a falling-off of fan efficiency would occur. The lowest permissible value of the product blade width  $\times$  blade velocity for good efficiency is of the order of magnitude of 2000 sq in. per second. Assuming a fan with a blade width of 5 in. at a hub of 15 in. diameter, this value would correspond to a fan turning speed of approximately 500 rpm. At this speed a fan wheel of present common design and, for instance, of 30 in. outside diameter, producing about 0.3 in. of pressure would have a noise level of less than 67 db (measured as an average of seven stations at 5 ft from the fan) if only inherent aerodynamic occurrences in the fan are considered.

Future improvements in the noise level of axial flow fans will consist in the author's opinion: (1) in judicious consideration of fan dimensions;

<sup>9</sup> Zur Schallstärke des von schnell bewegten Profilen erzeugten Schalles, by Holle-Lübcke. (Noise Level for Rapidly Moving Sections—Luftfahrtforschung, Vol. 17, p. 56, 1940.)

(2) in avoidance as far as possible of all additions to the basically necessary noise level; (3) in a study of possible improvements of the sound pattern connected with the arrangement of revolving blades by proper distribution of the aerodynamic pressure pattern over the disc area; and, (4) in employment of sound deadening or damping means either in the form of noise absorbents or of properly chosen resonators.

It is difficult to guess the further possible reduction of the noise level of axial flow fans. The variables entering into consideration have as yet, in the

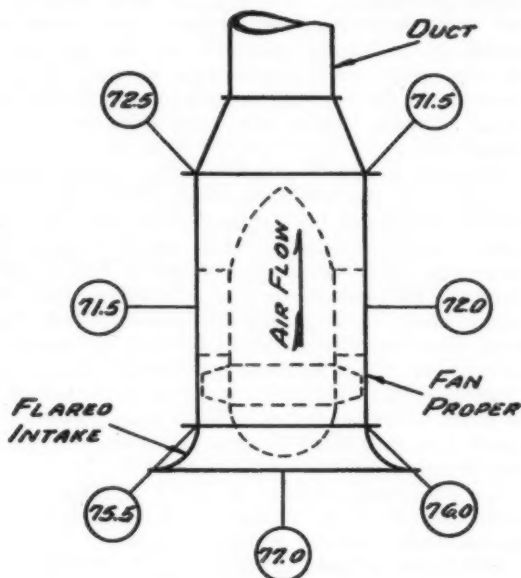


FIG. 11. COMPARATIVE NOISE LEVELS (IN DECIBELS)

author's belief, not been subjected to a complete systematic investigation along lines similar to those carried out for aeroplane propellers. Such an investigation should establish mathematical expressions describing the effect of variations in the static pressure pattern around the rotating fan disc similar to those used in the study of propeller noise. To this should also be added an investigation of the effect of the installation to which the fan itself is attached.

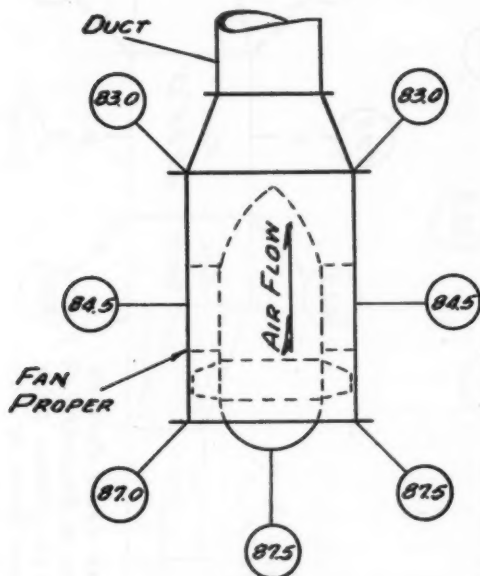
#### INSTALLATION OF FANS

The foregoing leads to the consideration of fan installation in general. Careful handling of the attachment of the fan to the connecting duct system will eliminate many sources of present complaint about fan noise and also sometimes of unsatisfactory air delivery.

Two examples of the effect of the installation on the resultant noise level are the following:

Fig. 11 shows noise level tests on a fan exhausting from a plenum chamber having a bell-shaped intake, and also on the same fan at identical conditions with a sharp-edged intake opening.

The noise reduction due to the flared intake will also depend on the size and shape of the flares. In one particular case the following variations were found in the



ABOUT A FAN WITH FLARED AND SHARP-EDGED INTAKES

noise for flares attached to a fan of  $15\frac{1}{2}$  in. diameter, when the flares were changed in size, while retaining the quarter-elliptic shape:

Max. flare dia./fan dia.	2.0	1.75	1.5	1.25
Noise level at position 5 ft ahead of fan.....	84.0 db	82.0 db	80.5 db	80.0 db
Average noise level (standard tests).....	78.8 db	78.2 db	77.7 db	77.3 db

This particular series of tests would indicate that a flared intake just large enough to assure reasonably smooth inflow to the fan is most favorable for a low noise level. Other available information, however, makes it questionable whether general use of flares as small as 1.25 times the fan diameter is advisable. Rigid installation rules cannot be given until a more complete analysis of the sound pressure field about a fan is available.

Fig. 12 shows a particularly striking example of an undesirable form of installation of a different nature from that of Fig. 11. It represents a fan

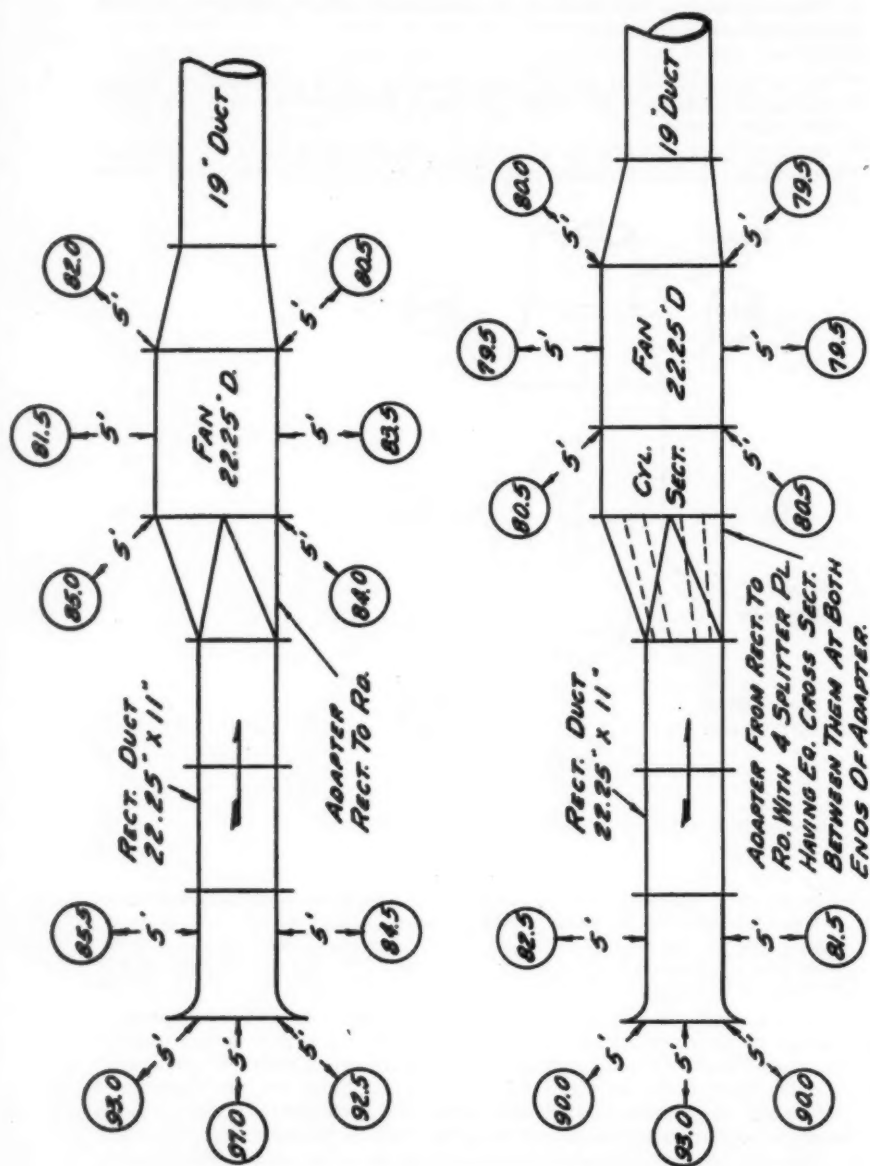


FIG. 12. NOISE LEVEL (IN DECIBELS) ON FAN INSTALLATION BEHIND A SHARP CHANGE IN DUCT CONTOUR

installed within a duct closely behind an abrupt change in duct cross section and alignment. This leads to an uneven flow of air into the fan wheel and related with this uneven flow, to a lowered performance and greatly increased noise. A complete remedy of the bad effects of the sudden change in direction and change in cross section on the fan performance cannot be given, but a partial relief can be obtained by securing a more even flow of air into the fan by means of the guide plate arrangement shown in the cut.

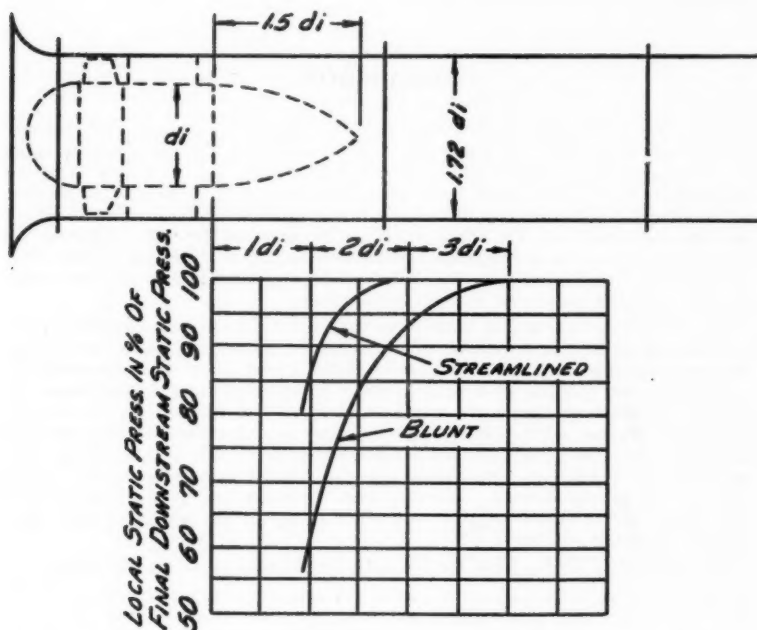


FIG. 13. DEVELOPMENT OF THE STATIC PRESSURE BEHIND AN AXIAL FLOW FAN WITH STREAMLINED AND BLUNT DOWNSTREAM COVER

In installing a fan caution also should be exercised to obtain the pressure for which the axial flow fan is rated. A fan having a streamlined tail as cover over the downstream side of its hub takes care of itself. Fans with blunt cut-offs in their center, however, in addition to having lower efficiency, also require a length of straight duct downstream for the build-up of the static pressure credited to them by standard test procedure. The test results given in Fig. 13 illustrate the conditions for an average case of this type. Also, a tail-less fan blowing into a plenum chamber immediately adjacent to the downstream fan end should not be expected to build up the pressure measured for fans installed in a long straight tube.

## CONCLUSION

The axial flow fan is essentially based on the aerodynamics of airfoils. Its performance may be further improved by a still better understanding of the forces developed on airfoils, by increased knowledge regarding the details of the pressure and of the sound field created by the rotating airfoils, and by the utilization of improved sections in the design of fan blades. Careful study of the fan installation will make certain that the fan operates as well as its test-stand performance would indicate.

## DISCUSSION

A. E. CRIQUI,\* Buffalo, N. Y. (WRITTEN): The author has given a presentation of many of the details in the design of axial flow fans. The use of aerodynamics in aeronautical work usually assumes free flow conditions, that is, a body immersed in an infinite fluid. When these same concepts are applied to fan design, the problem is complicated by the confining boundaries of the fan. The mutual effect of blades must be considered, also the effect of the fan housing and fan hub. Likewise for the guide vanes, it is not enough to consider them as highly cambered airfoils. The full fan performance curve from shutoff to free delivery takes in a much wider range of angle of attack than used in aeronautical work.

The author's opinion is that appreciable volume control can be accomplished only by adjusting the pitch of the blades. There is another means in extensive use on high pressure axial flow fans. This is by using variable inlet guide vanes, controlled by a damper motor. These vanes in one setting act as inlet vanes. They rotate the air in a direction opposite to the wheel rotation and thus increase the normal performance of the fan. When these vanes are rotated to a closed position they not only reduce the air volume, but the fan horsepower as well. With this arrangement, capacities are reduced to considerably less than 50 per cent. Horsepower too is reduced a comparable amount. The fans under discussion are large size, running at high speed. By keeping the control on a non-rotating part in this case a very satisfactory solution is obtained.

The constant in the fan noise laws is fairly well agreed on as the following:

$$\text{db change} = 50 \log_{10} \frac{\text{RPM}_2}{\text{RPM}_1} \quad \text{or} \\ 50 \log_{10} \frac{\text{tip speed}_2}{\text{tip speed}_1}$$

It is to be noted that this law assumes constant rating point and constant size. This law has been accepted by the *National Association of Fan Manufacturers* and is given in their Bulletin No. 104—Sound Measurement Test Code for Centrifugal and Axial Flow Fans.

This code qualifies the law by limiting its application to a range of 10 per cent speed change. One reason for so doing is to allow for noise components, as the author states, not inherently connected with the fan action. An example of such an extrinsic source might be the motor in the case of a two-speed fan. Assuming an abnormally loud motor, and a quiet fan, at the low speed the motor might produce a sizeable portion of the entire unit noise. At high speed the motor noise, if it did not increase materially, might be masked by the predominant fan noise. As a result, the noise level of the unit would not increase as much as the constant 50 would

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indicate. For the average carefully designed fan, test data substantiate the law for ranges much more than the 10 per cent mentioned at beginning of this paragraph.

R. T. LANGE, Piqua, Ohio: During the past 15 years I have done quite a bit of testing on propeller type fans and have just recently completed about 10 months of intensive testing on a small 7 in. diameter axial flow fan, which could actually be called a modified screw pitch propeller fan. The results have been very interesting. There is an application for a fan of this type in the aviation industry where it is used in conjunction with a gas fired type of unit heater.

I have found it to be generally true that whenever one has investigated the test data presented on a new type or design of fan, that the test data has not been obtained in a manner which conforms or is in accord with the society's standard fan testing code. Usually when tested according to our code the efficiencies are found to be lower than those obtained by the non-standard method of testing.

It may be of interest to some of those present that the fan which I have just completed testing was run at speeds as high as 12,000 rpm. The unit was run as a single, two-, three- and four-stage assembly. Tests were also made with a different number and various types of guide vanes in the assembly. Static efficiencies varied from 58 per cent to 87 per cent for peak values.

Since the unit tested was 7 in. O.D. it was not difficult to proceed with testing strictly in accord with the code as the test setup did not require much space.

My question to Messrs. Criqui and Heath,<sup>10</sup> and to the author of this paper is whether the data which they have presented was obtained by the Society's standard code<sup>11</sup> method for testing fans.

MR. TROLLER: The answer is-Yes, they were made under standard test conditions.

R. D. MADISON, Buffalo, N. Y.: In reference to a question as to how such high efficiencies could be obtained by axial flow fans, I worked out in diagram form and can show how the space between airfoil blades could be likened to a curved diverging channel. In duct work it is difficult to get air to flow efficiently in such a channel. However, the air passing around each airfoil shape attains a circulation component which furnishes the turning movement required. Thus what is inefficient flow in a single diverging elbow may become a very efficient flow through the blades of an axial flow fan.

HARRY SCHMIDT, Caldwell, N. J.: In Mr. Heath's paper<sup>12</sup> I noted that the characteristic curves were all given at a constant speed. The authors bring out the different characteristics and efficiencies of fans at the different parts of the curve. The blade is designed in an attempt to obtain a medium characteristic so as to obtain the best efficiency. The author of this paper recognized the fact in the thought that he would like to see a variable pitched blade fan which would overcome the inefficiencies that are obtained in the axial flow fan and make it uniformly efficient throughout its characteristic curve.

Those familiar with the variable pitch airplane propeller know what an intricate mechanism this is, and what it would involve in the way of cost. Therefore, it seems to be impracticable. It may be practicable in Europe, where labor costs are relatively cheap, but here in commercial work I do not think it will be obtainable.

One factor has been overlooked in attaining the ideal axial flow fan and that is speed regulation. The attempt has been made by varying the inlet dampers or inlet vanes or exit vanes, and the fact has been lost that there will be cheap means of speed variation available after the war, in which motors can be obtained either electrically controlled or built-in type, that will give us all the characteristics of

<sup>10</sup> The Axial Flow and Its Place in Ventilation, by W. R. Heath and A. E. Criqui, see p. 197, this volume.

<sup>11</sup> Standard Test Code for Centrifugal and Axial Fans (Bulletin 103), published by National Association of Fan Manufacturers, Detroit, Mich.

<sup>12</sup> Loc. Cit. Note 10.

the standard direct current motor, and therefore, you will be able to obtain the performance desired from the axial flow fan.

The survey of noise is one thing that is interesting in this paper. I notice that the author did not give any velocity contours of the air entering the fan. Such contours would be a good guide to the cause of the loud noise effect of the axial flow fan.

MR. TROLLER: I should like to say, first, that, in judging the advantages of adjustable pitch, we start with the opinion that it will be possible, in the particular applications we are confronting in the ventilation industry, to build axial flow fans with adjustable pitch of these various degrees of changing the blade pitch at a reasonable cost. Beyond that, I would like to say that, theoretically, a number of the changes in characteristics could be obtained by means of speed change, but that the reason for not using speed change is that so far no good means are available for changing AC motor speed. If improved means of obtaining speed change become available, then I am sure the question will simply be that of determining which one of the two means is the better in each individual case.

W. H. CARRIER, Syracuse, N. Y.: I believe it is of interest in this discussion to know what is being done with an axial flow fan on a wind tunnel at Cleveland. This fan handles a little over 10,000,000 cu ft of air per minute at about 10½ in. static pressure with a direct connected 20,000 hp motor. The expected efficiency is between 85 per cent and 87 per cent.

Y. S. TOULOUKIAN, West Lafayette, Ind.: My question is addressed to the author. He mentioned this boundary layer problem in only one of his figures. Would the author care to comment on the effect of the boundary layer in eliminating the noise, if any?

AUTHOR'S CLOSURE: By eliminating the boundary layer by means of this suction control, as it is known in aerodynamics, it becomes possible to use higher pressures—that is, higher lift forces on a given blade at a given speed. If such higher pressures are used then, of course, it may be possible to obtain a given static pressure requirement from a fan at lower speed. By this combination of using lower speeds you may, under certain circumstances, arrive at a resultant over-all noise lower than that with conventional fans. The existing noise level in a conventional fan is partly to be traced to the existing pressure field about the individual blade, to the relative pressure changes and to the total velocity existing at these pressure coefficients. By having at a given velocity available higher pressures on one blade, better resultant combinations of pressure field and noise level are at least within the range of possibility.



**1252**

## OPTIMUM SURFACE DISTRIBUTION IN PANEL HEATING AND COOLING SYSTEMS

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Second progress report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of California.

### INTRODUCTION

ANY HEATING system designed to provide occupant comfort by control of radiant energy exchange must, to be effective, satisfy three fundamental conditions. These are:

1. To relate the mean radiant temperature (MRT) of the enclosure to the air temperature ( $t_a$ ) in such a way that the comfort equation will be satisfied.
2. To utilize a maximum irradiation, Btu/(hour) (sq ft of occupant), less than that at which discomfort is first experienced.
3. To distribute the basic radiating surface in a manner such that direct irradiation of the occupant will be satisfactorily uniform for all parts of the conditioned enclosure.

Applied to the design of a panel heating system, the criteria given require determination of the minimum panel area needed to maintain an adequate MRT under conditions of maximum heat loss; the selection of a maximum panel temperature sufficiently low so that the intensity of radiation from the panel will not be objectionable; distribution of the design panel area in such a manner as will avoid wide divergence in the irradiation of occupants as they move about in the room. A radiant heating system for which the second and third of these criteria are often critical is the standard residential fireplace. The high flame temperature gives an intensity which is objectionable to anyone seated near the fire while the distribution of radiating surface is so concentrated that uniformity of heating is likely to be very poor. Under such conditions an occupant may be in a position for which a satisfactory over-all heat balance is established on his body, yet one part of the body will be subject to discomfort arising from extreme irradiation while, simultaneously, another part is uncomfortably cold.

Except in the case of incandescent heating surfaces, the intensity problem does not arise in designing a radiant heating system. Panels of conventional design operate in a temperature range (80 to 180 F for heating and 50 to 70 F for cooling) in which the intensity is not outside the comfort range

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and therefore need not be given consideration by the designer. Because of this fortunate fact, the designer's attention is sometimes distracted from the unfortunate fact that *uniformity* is a factor of genuine practical importance and one which requires careful consideration if the system is to provide comfort irrespective of the position of the occupant in the enclosure. Fundamentally, the procedure in designing a panel system is first to select a design maximum panel surface temperature, then calculate the required panel area and, lastly, distribute this area of panel in a manner—determined from the geometry of the heated space—such that uniformity will be attained. The first and second steps in panel design have been discussed in earlier papers.<sup>1</sup> The intent of the present paper is to provide basic design data and indicate a rational procedure for carrying out the third necessary step in the design of a comfort panel heating (or cooling) system.

The fact should constantly be remembered that in the design step having to do with uniformity, the absolute value of the energy exchange between the panel and the occupant is not of consequence; interest is centered on the percentage variation occasioned by movement of the occupant in the room (or by change in position of the occupant when remaining at a fixed point in the room). The most effective design will, in every case, be the one for which the per cent variation is a minimum. The absolute value of the radiant energy exchange between occupant and panel will vary, for the same geometric distribution of panel surface, as a function of the total panel area, but this variation is inherent in the heat transfer characteristics of the particular system and is, therefore, not a factor amenable to treatment from the standpoint of distribution design. Distribution, fundamentally, is fixed in terms of the space relationships between the occupants and the panels; for a room of fixed plan and ceiling height there is, therefore, a particular panel pattern which, irrespective of panel design temperature or of panel area, will give greatest uniformity of direct exchange between occupant and primary radiating (or, for panel cooling, receiving) surface. Thus the problem of designing for uniformity of heating effect reduces to one of geometry rather than heat transfer and permits realization of a series of geometrical solutions which are applicable to systems having any heat loss characteristics.

This generalization of the distribution problem enormously simplifies the work of the designer, since it permits solution of the two basic design problems independently of one another. A knowledge of the ventilation and heat transfer characteristics of the structure permits determination of the necessary panel area (operating at design temperature) without consideration of the shape of room or location of panels. Consideration of the geometry of the system permits determination of the optimum panel pattern. Distribution of known panel area in a known pattern is then a matter of simple arithmetic to fix the width of panel per lineal foot of pattern.

Although knowledge of the absolute radiant energy exchange between panel and occupant is unimportant in conventional panel heating systems, there are special installations for which such information may be of great importance. In the conventional system, as for a residence or office building,

<sup>1</sup> Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941.)

Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942.)

Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942.)

the primary purpose of the panel is to establish the desired MRT; this is accomplished largely by radiant exchange between walls and panel; the direct energy flow from panel to occupant is consequently of only minor importance. In many special cases, however, the use of panels to raise the MRT of the entire enclosure may be impractical or uneconomical and in such instances direct energy transfer to or from the occupant can be resorted to as a method of establishing localized or *spot* heating or cooling. Systems of this kind find use in industry for marine installations (as in engine rooms or galleys) and seem also to offer intriguing possibilities for use in the cabins and cockpits of airplanes intended for high altitude operation. Design of such installations requires knowledge of the absolute exchange rate of energy between panel and subject; fortunately the same basic experimental data (reported in this paper) which are necessary for solving the problem of uniform energy distribution are also adequate for use in calculations of absolute exchange rates.

#### EXPERIMENTAL PROCEDURE

Distribution of panel area for optimum uniformity of heating requires knowledge of the variation in direct radiant energy exchange between the occupant and the panel as the occupant varies his position with respect to the panel. Since the exchange rate, for a fixed position of the occupant, is the sum of the exchange rates for all unit areas of the panel, the problem resolves itself into a determination of the exchange variation between a fixed unit area (or infinitesimal area) of panel and a moving occupant. More simply, the experimental data can be collected for a system in which the occupant retains a fixed position and the element of panel is moved. If the human body were cylindrical the experimental procedure would reduce, for a fixed ceiling height, to determination of exchange between the cylinder and an elementary ceiling area as that area moved radially outward from the center line of the cylinder. But the human form markedly differs from that of a cylinder and it is therefore necessary to investigate exchange rates not only for radial variation of the elementary area but also for variation circumferentially. For a subject in a sitting position experimental determinations are necessary through 180 deg, but for a standing subject experiment showed that no material difference in exchange rates occurs outside of that included in the 90 deg range from full face to full profile. Further, the circumferential variation was found to proceed without marked discontinuity so that the experimental procedure could be reduced to determinations for three angles for the standing figure (full face, 45 deg off of full face, full profile) and for only two additional angles (full rear and 45 deg off of full rear) for the seated figure.

Assuming that the panel is a diffuse surface source, exchange rate variation is a function of the cosine of the angle,  $\phi$ , between a normal to the panel surface and a line connecting the center of the elementary emitter and receiver areas (see Appendix for equations) and is also a function of the projected area of the receiver on a plane normal to the line connecting emitter and receiver; by the inverse square law the shape factor is also an inverse function of the length of this line squared. For an elementary panel area the exchange rate variation (as a function of subject position) thus depends on integration, over the area of the subject *seen* by the elementary panel area, of the product  $\cos \phi$  by the projection of the corresponding

elementary area of the subject and the inverse of the distance squared. Evaluation of this integral by analytical methods is impractical for a receiver surface as complex as that represented by the human body. Thus the need arises for experimental integration by laboratory means.

The data presented in this paper were obtained by use of a mechanical integrator of a type proposed by Hottel<sup>2</sup> and developed by Boelter;<sup>3</sup> a view

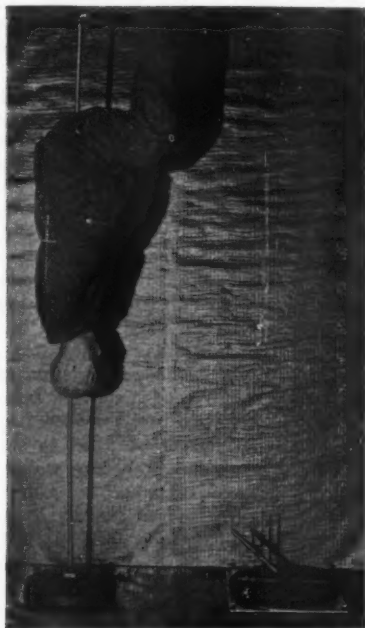


FIG. 1. DUMMY SUBJECT IN SEATED POSTURE AND THE MECHANICAL INTEGRATOR

of this instrument appears in lower right, Fig. 1 (see scale drawing Fig. 1a). As used in the laboratory, the integrator was placed directly over the point selected to represent the center of the elementary panel area. From this position a beam of light from the integrator was directed around the outline of the subject. For each such determination the integrator drew a closed curve on a piece of paper; the area within the curve was determined with a planimeter and divided by a constant to obtain the shape factor which is equal to the fraction of energy received by the subject of that emitted by the elementary panel.

Convenience in operation of the integrator required greater accessibility than could have been obtained if the integrator had been moved about on the

<sup>2</sup> Radiant Heat Transmission, by H. C. Hottel, *Mechanical Engineering*, Vol. 52.

<sup>3</sup> A Mechanical Integrator for the Determination of the Illumination from Diffuse Surfaces Sources, by V. H. Cherry, D. D. Davis, L. M. K. Boelter, *Transactions, Illuminating Engineering Society*, 1939.

ceiling. Therefore it was decided to *invert* the system, placing the subject in a head down position with feet on or near the research room ceiling and moving the integrator from point to point on the floor of the room. An additional advantage of this inverted system (see Fig. 1) is that it permitted simple variation in *ceiling* height; this height being, in each case, the vertical distance from the feet of the dummy to the floor of the research room.

Concern was also felt for the variations in body size among subjects and variation in form between the male and female figure. No attempt has been made to take account of such variations; the data presented are, without

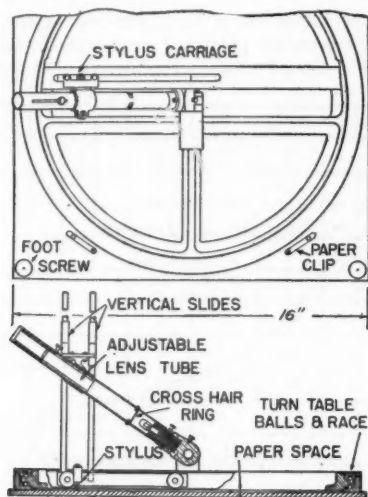


FIG. 1A. THE MECHANICAL INTEGRATOR

exception, taken from an *average* male subject of height 5 ft 10 in. and weight 165 lb. From a study of the experimental shape factors for such a figure the effect of change in height can be estimated.

#### EXPERIMENTAL RESULTS

**Ceiling Panels.** The first group of tests were conducted for a *standing* subject located in a room with ceiling panels. Ceiling heights of 8 ft, 10 ft, and 12 ft were investigated. Fig. 2 gives the results for an 8-ft ceiling. For a fixed position of the subject the three curves of this figure give the variation in fraction of energy which reaches the subject from a panel of elementary area as it is moved out from the vertical center line of the subject. The abscissa can, of course, be considered to represent either the horizontal distance of unit panel from a fixed subject or of subject from a fixed unit panel.

Two items of particular interest are evident from this figure:

1. Even with a low (8 ft) ceiling, the maximum fraction of energy leaving a panel element and reaching an occupant of a panel-heated room is less than 5 per cent. The

fraction contributed by heated surface situated more than 6 ft radially out from a subject is less than 2 per cent while for radial distances exceeding 8 ft less than 1 per cent energy leaving the panel passes directly to the occupant.

2. Maximum exchange does not occur when the occupant is directly under the elementary area. When in this position the panel *sees* only the head and shoulders of the subject and the consequent reduction in projected area exceeds the advantage due to a smaller average angle  $\phi$ . As the subject moves out from directly under the elementary panel area a rapid increase in projected body area takes place until, for a radial distance of from  $1\frac{1}{2}$  ft to 2 ft, the maximum exchange rate is realized. For distances beyond 2 ft the projected area continues to increase for a short interval and then rapidly decreases with distance, but even in the short interval the product of projected area,  $\cos \phi$ , and the inverse of distance squared is found to decrease.

Thus the typical shape factor curve for a standing figure is in three distinct parts; in the first part increasing projected area controls and the shape factor becomes greater; in the second part increasing area is more than offset by

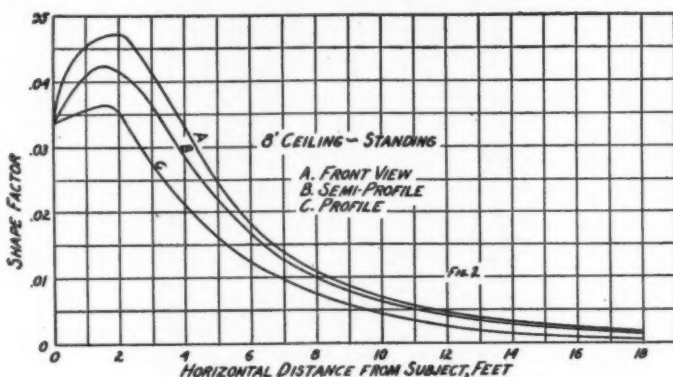


FIG. 2. FRACTION OF ENERGY REACHING A STANDING SUBJECT FROM AN ELEMENTAL PANEL AREA IN AN 8 FT CEILING

rapidly increasing  $\cos \phi$  and increasing distance; in the last part, the changes in projected area and in  $\cos \phi$  are less significant than the effect of distance.

A three-dimensional solid, the height representing shape factor, would be obtained if the curve for profile were rotated 90 deg with respect to the plane of the page (Fig. 2) and the curve for semi-profile were rotated 45 deg with respect to the same plane. The slope of the surface for such a solid would approach variation by the inverse square law for radial distances greater than approximately 12 ft. Note that at a distance of 18 ft out from the elementary area, the exchange rate has dropped to approximately one-eighth of its maximum; thus a variation of over 800 per cent occurs in direct heating effect as one moves out 18 ft from an elementary ceiling panel located in a room of 8 ft height.

Fig. 3 is for a standing subject with ceiling panels in a room having a 10 ft ceiling, while Fig. 4 is for a similar subject in a room with 12 ft ceiling. The radial distance for maximum exchange is seen to increase from  $1\frac{1}{2}$  ft to 3 ft to 5 ft as the ceiling height increases from 8 ft to 10 ft to 12 ft. Absolute energy exchange rate falls off very rapidly with ceiling height so that for a 12 ft ceiling only one-half of one per cent of the energy leaving

the elementary panel area is received by a subject standing directly below, while at 18 ft the amount received is less than one-quarter of one per cent.

The second group of ceiling panel tests were for a subject in *seated position* in rooms with ceilings of 8 ft, 10 ft, and 12 ft; data for these cases are given in Figs. 5, 6, 7, respectively. The previous discussion of the data for a standing subject is also applicable to these three sets of curves with the addition of a note concerning a peculiarity of the curve for back view of a seated figure in a room having 8 ft ceiling height. For this one case it is observed that a reflex occurs in the curve as the subject moves radially outward from a

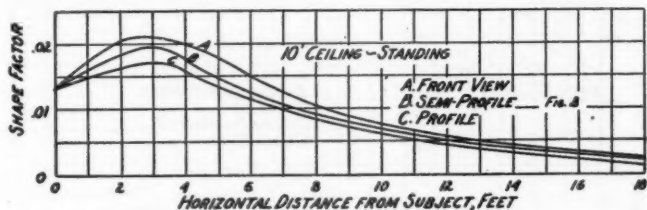


FIG. 3. FRACTION OF ENERGY REACHING A STANDING SUBJECT FROM AN ELEMENTAL PANEL AREA IN A 10 FT CEILING

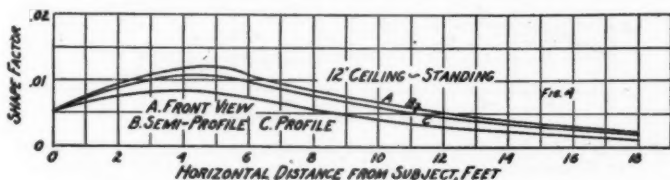


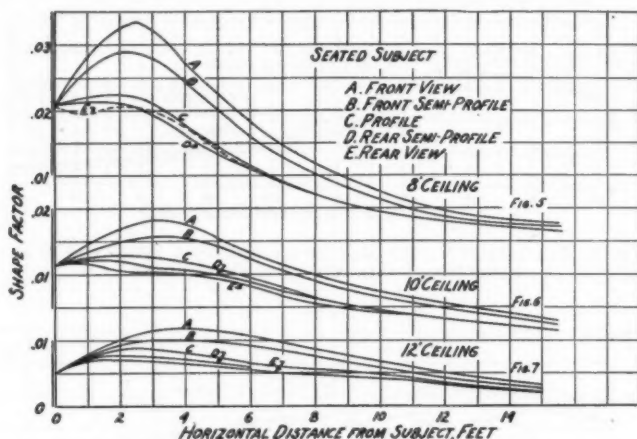
FIG. 4. FRACTION OF ENERGY REACHING A STANDING SUBJECT FROM AN ELEMENTAL PANEL AREA IN A 12 FT CEILING

position directly under the panel. This characteristic results from the fact that, when directly under the panel, the thighs and knees of the seated subject contribute to the projected area; as radial movement occurs this section of the subject's body is rapidly shielded by his head, shoulders and back so that, for a brief interval, the increase in projected area due to exposure of the back is more than offset. Soon, however, the thigh area is completely shielded; increasing exposed back area then causes the shape factor to increase for a short radial distance before the usual maximum and subsequent reduction sets in. For the 10 ft and 12 ft ceilings this inversion of the curve is not noted though some evidence of the *thigh effect* does appear in two of the curves of Fig. 6.

Comparing Figs. 2 and 5 shows that, for the same ceiling height, radiant exchange to a subject directly under an element of panel is much greater when the subject is standing than when he is sitting. At first glance this may seem surprising since the projected body area is greater for the seated subject,

but the greater average distance of the seated subject more than offsets the increased area. At radial distances greater than 6 ft, the differences in exchange for the seated and standing positions are practically negligible.

**Wall Panels.** The variation in shape factor of the subject with respect to an elementary area of wall panel is very much more difficult to evaluate than for ceiling panels. For this case the experimental work was limited to investigation of vertical wall panels located in a plane making a 90 deg horizontal angle (90 deg in plan view of the room) with the straight line connecting the element of area to the feet of the standing subject (see Fig. 8). Data



FIGS. 5, 6, 7. FRACTION OF ENERGY REACHING A SEATED SUBJECT FROM AN ELEMENTAL PANEL AREA IN AN 8 FT, 10 FT, OR 12 FT CEILING

were obtained for full face and for profile positions of the subject with respect to the panel and for elements of area located at horizontal distances of 2 ft, 4 ft, 6 ft, 8 ft, 10 ft, and 12 ft from the subject and at 2 ft, 4 ft, 6 ft, 8 ft, 10 ft, and 12 ft distances above the floor. The results of the experimental work are shown in Figs. 9 and 10.

Extension of the experimental data to cases where the subject faces an element of panel which is in a plane having a normal which makes a horizontal angle  $\beta$  with a straight line connecting the element and the feet of the subject (see Fig. 8) can be readily accomplished by obtaining the shape factor from the experimental curves for an equivalent distance  $L$  equal to the actual distance  $L$  ft and multiplying this factor by  $\cos \beta$ . Thus, referring to Fig. 8, the shape factor at any point  $X$  is given in terms of the experimentally determined shape factor at  $P$  by the equation,

$$F_x = F_P \cos \beta = F_P \frac{L}{\sqrt{L^2 + D^2}}$$

Shape factors for the range of values of  $D$ ,  $H$ , and  $L$  which are of practical significance have been computed by Charles F. Dischler and are presented in

Tables 1 and 2 for full face and profile positions, respectively. From these tabular point values the over-all shape factor of a subject with respect to a wall panel of any size and shape can be obtained by summing the tabular shape factor at mid-points of small equal area elements of the panel and then dividing the sum by the number of elements so considered. This method does not give an exact result because it does not permit taking into account the variation from full face (or from profile) of the figure, but the inaccuracy is unlikely to be of practical significance and can, if necessary, be reduced by interpolating for each value of  $D/L$  between the corresponding tabular values for full face and profile positions.

*Floor Panels.* The maximum operating temperature of floor panels is so much lower than for ceiling panels and the fraction of convective heat transfer

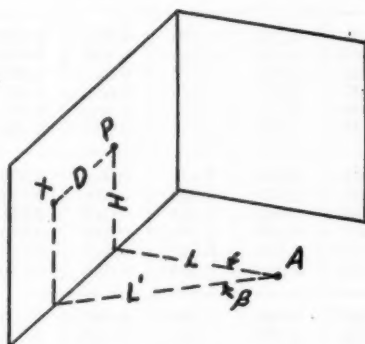


FIG. 8. THE FLOOR ANGLE  $\beta$  WITHIN WHICH WALL PANELS AS AT X WERE CONSIDERED; SUBJECT WITH FEET AT A

so much greater, that direct irradiation of the subject in such a panel system is likely to be insignificantly small. For this reason the experimental work necessary for obtaining shape factor data for floor panels could not be justified. Inspection of Figs. 2, 3, and 4 for ceiling panels reveals, however, that the non-uniformity of primary irradiation varies inversely at ceiling height. Since a floor system is approximately equivalent to a ceiling system in a room having 6 ft ceiling height (the degree of the approximation being represented by the difference in shape of a subject standing normally from one standing on his head) it appears evident that reasonable uniformity of direct irradiation from floor panels would be practically impossible of attainment. Fortunately the low temperature and small direct irradiation fraction in floor systems makes this non-uniformity less important than in wall and ceiling systems where greater departures from the characteristics of conventional convective systems is attained.

#### OPTIMUM PANEL DISTRIBUTION FOR UNIFORMITY OF PRIMARY IRRADIATION

*Calculation of Absolute Shape Factors.* The experimentally determined basic shape factor data have two distinct fields of usefulness. The first is in cal-

TABLE 1—SHAPE FACTORS OF SUBJECT IN FULL FACE WITH RESPECT TO ELEMENT OF WALL PANEL

 $L$  = length of normal from wall to feet of subject $H$  = height of panel element $D$  = horizontal distance of panel element to right or left of normal from wall to feet of subject

SHAPE FACTORS								
$L$	$D$ —FT	$H$ —FT						
		0	2	4	6	8	10	12
2 ft	2	0.108	0.211	0.230	0.094	0.020	0.005	0.002
	4	0.068	0.133	0.145	0.060	0.012	0.003	0.001
	6	0.048	0.094	0.103	0.042	0.009	0.002	0.001
	8	0.037	0.072	0.079	0.034	0.007	0.002	0.001
	10	0.030	0.058	0.064	0.026	0.006	0.002	0.001
	12	0.025	0.049	0.054	0.022	0.005	0.001	0.001
4 ft	2	0.064	0.098	0.101	0.064	0.029	0.013	0.006
	4	0.050	0.077	0.080	0.051	0.023	0.010	0.005
	6	0.039	0.061	0.063	0.040	0.018	0.008	0.004
	8	0.032	0.049	0.051	0.033	0.014	0.006	0.003
	10	0.026	0.041	0.042	0.027	0.012	0.005	0.002
	12	0.022	0.034	0.036	0.023	0.010	0.004	0.002
6 ft	2	0.038	0.048	0.049	0.040	0.025	0.014	0.008
	4	0.034	0.042	0.042	0.035	0.022	0.012	0.007
	6	0.029	0.036	0.036	0.030	0.019	0.010	0.006
	8	0.024	0.031	0.030	0.026	0.016	0.009	0.005
	10	0.021	0.026	0.026	0.022	0.014	0.007	0.004
	12	0.018	0.023	0.023	0.019	0.012	0.007	0.004
8 ft	2	0.024	0.026	0.025	0.025	0.019	0.013	0.009
	4	0.022	0.024	0.023	0.023	0.018	0.012	0.008
	6	0.019	0.021	0.021	0.021	0.016	0.011	0.007
	8	0.017	0.019	0.018	0.018	0.014	0.010	0.006
	10	0.015	0.017	0.016	0.016	0.012	0.009	0.006
	12	0.014	0.015	0.014	0.014	0.011	0.008	0.005
10 ft	2	0.016	0.016	0.015	0.017	0.014	0.011	0.008
	4	0.015	0.015	0.014	0.016	0.014	0.011	0.007
	6	0.014	0.014	0.013	0.014	0.013	0.010	0.007
	8	0.013	0.013	0.012	0.013	0.012	0.009	0.006
	10	0.012	0.011	0.011	0.012	0.010	0.008	0.006
	12	0.011	0.010	0.010	0.011	0.009	0.007	0.005
12 ft	2	0.011	0.011	0.011	0.012	0.011	0.009	0.007
	4	0.011	0.011	0.011	0.012	0.011	0.009	0.007
	6	0.010	0.010	0.010	0.011	0.010	0.008	0.007
	8	0.010	0.010	0.010	0.010	0.010	0.008	0.006
	10	0.009	0.009	0.009	0.009	0.009	0.007	0.006
	12	0.008	0.008	0.008	0.009	0.008	0.007	0.005

TABLE 2—SHAPE FACTORS OF SUBJECT IN PROFILE WITH RESPECT TO ELEMENT OF WALL PANEL

 $L$  = length of normal from wall to feet of subject $H$  = height of panel element $D$  = horizontal distance of panel element to right or left of normal from wall to feet of subject

SHAPE FACTORS								
$L$	$D$ —FT	$H$ —FT						
		0	2	4	6	8	10	12
2 ft	2	0.061	0.139	0.144	0.059	0.013	0.004	0.002
	4	0.039	0.088	0.091	0.037	0.008	0.002	0.001
	6	0.027	0.062	0.065	0.026	0.006	0.002	0.001
	8	0.021	0.049	0.050	0.020	0.004	0.001	0.001
	10	0.017	0.039	0.040	0.016	0.004	0.001	0.001
	12	0.014	0.032	0.034	0.014	0.003	0.001	0.0004
4 ft	2	0.033	0.051	0.053	0.035	0.018	0.008	0.003
	4	0.026	0.041	0.042	0.028	0.014	0.006	0.003
	6	0.020	0.032	0.033	0.022	0.011	0.005	0.002
	8	0.016	0.026	0.026	0.018	0.009	0.004	0.002
	10	0.014	0.021	0.022	0.015	0.007	0.003	0.001
	12	0.012	0.018	0.019	0.013	0.006	0.003	0.001
6 ft	2	0.020	0.025	0.026	0.022	0.016	0.009	0.005
	4	0.018	0.022	0.022	0.019	0.014	0.008	0.004
	6	0.015	0.019	0.019	0.016	0.012	0.006	0.004
	8	0.013	0.016	0.016	0.014	0.010	0.005	0.003
	10	0.011	0.014	0.014	0.012	0.009	0.005	0.003
	12	0.010	0.012	0.012	0.010	0.007	0.004	0.002
8 ft	2	0.013	0.014	0.014	0.014	0.012	0.008	0.005
	4	0.012	0.013	0.013	0.012	0.011	0.007	0.005
	6	0.011	0.012	0.011	0.011	0.010	0.006	0.004
	8	0.010	0.010	0.010	0.010	0.009	0.006	0.004
	10	0.009	0.009	0.009	0.009	0.008	0.005	0.003
	12	0.008	0.008	0.008	0.008	0.007	0.004	0.003
10 ft	2	0.009	0.010	0.010	0.011	0.009	0.006	0.004
	4	0.008	0.010	0.009	0.010	0.008	0.006	0.004
	6	0.008	0.009	0.008	0.009	0.008	0.006	0.004
	8	0.007	0.008	0.008	0.008	0.007	0.005	0.004
	10	0.006	0.007	0.007	0.008	0.006	0.005	0.003
	12	0.006	0.007	0.006	0.007	0.006	0.004	0.003
12 ft	2	0.006	0.006	0.006	0.007	0.006	0.005	0.004
	4	0.006	0.006	0.006	0.006	0.006	0.005	0.004
	6	0.006	0.006	0.006	0.006	0.006	0.005	0.003
	8	0.005	0.005	0.005	0.006	0.005	0.004	0.003
	10	0.005	0.005	0.005	0.005	0.005	0.004	0.003
	12	0.005	0.005	0.004	0.005	0.005	0.004	0.003

culating the over-all shape factor of a subject in given position with respect to a panel of any shape and size. For this purpose the plan view of the panel is drawn to scale and the panel then arbitrarily divided into small equal area elements; the smaller the element of area the greater will be the accuracy of the calculated result, but for practical design purposes there is never need

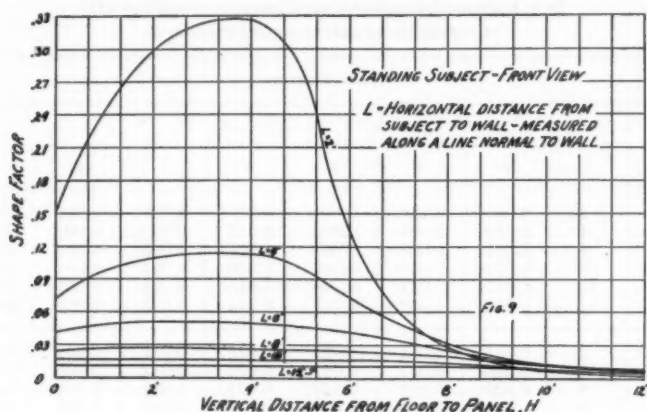


FIG. 9. FRACTION OF ENERGY REACHING FRONT OF STANDING SUBJECT FROM AN ELEMENTAL PANEL AREA IN A WALL

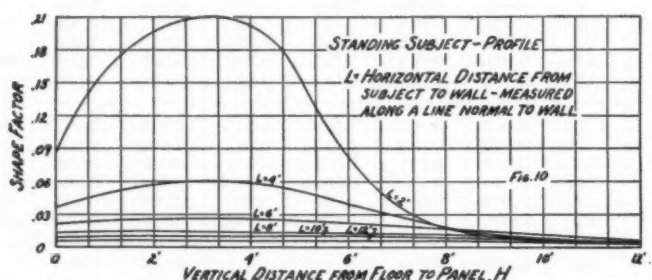


FIG. 10. FRACTION OF ENERGY REACHING SIDE OF STANDING SUBJECT FROM AN ELEMENTAL PANEL AREA IN A WALL

for taking elements less than one square foot and in many cases squares two feet on the side will give results of reasonable accuracy. The location of the subject is fixed on the plan view and the horizontal distance from subject to center of each elementary area is then scaled. For a room of known ceiling height and for a particular facing direction the shape factor of subject with respect to each unit of area can be determined, for the scaled horizontal distance, by interpolation between the shape factor curves for profile, semi-profile and full face. Adding the factors so determined and dividing by the

number of equal area elements gives the average unit area shape factor which is numerically identical with the shape factor of the subject with respect to the entire panel.

A numerical example of such an absolute shape factor determination is illustrated in Fig. 11. The subject is standing at point *P* in a room 11 ft x 11 ft with 8 ft ceiling height. The entire ceiling is acting as a heating panel at uniform surface temperature and the shape factor of subject with respect to ceiling is desired. As shown in the figure, the ceiling is divided into elementary areas of 1 sq ft each. The distance from center of each such area to the subject is then scaled and the corresponding shape factor obtained from Fig. 2 and recorded in its proper square. The variation from full face to profile position was neglected in this example and all data recorded for a

.039	.091	.042	.037	.089	.083	.017	.013	.010	.008	.007	
.041	.093	.040	.035	.085	.081	.015	.011	.008	.006	.005	
.042	.090	.037	.032	.085	.080	.015	.010	.008	.005	.004	
.037	.085	.032	.027	.080	.079	.014	.009	.007	.004	.003	
.029	.083	.028	.022	.081	.078	.012	.008	.006	.004	.003	
.023	.081	.020	.018	.075	.070	.009	.007	.005	.004	.003	
.017	.075	.015	.010	.065	.060	.007	.005	.004	.003	.002	
.013	.070	.012	.011	.060	.055	.007	.005	.004	.003	.002	
.010	.065	.009	.009	.055	.050	.006	.005	.004	.003	.002	
.008	.060	.007	.007	.050	.045	.005	.004	.003	.002	.002	
.007	.055	.006	.006	.045	.040	.004	.004	.003	.002	.002	

FIG. 11. ABSOLUTE SHAPE FACTOR DETERMINATION FOR A SUBJECT AT *P* IN A ROOM 11 FT X 11 FT WITH AN 8 FT CEILING

semi-profile view of the subject. Except in cases where precise investigation for some special purpose is required, the assumption of a semi-profile view seems to be a reasonable average approximation for all cases and will be used as such in all numerical examples given in this paper. For the case now being considered the required shape factor is equal to  $1/121$  of the sum of all unit area factors; the result is thus,  $1.7275 \div 121 = 0.0143 = 1.43$  per cent, that is, slightly less than one and a half per cent of the total energy leaving the ceiling is directly received by the subject.

As a second example consider a heating surface made up only of the shaded sections in Fig. 11; the shape factor for such a panel is, again, the sum of factors for all heated elements divided by the number of such elements  $1.0050 \div 68 = 0.0148 = 1.48$  per cent. In a similar manner Fig. 11 will provide data to calculate the shape factor of the subject, when at *P*, with respect to any shape of panel. A further usefulness of the data from this figure arises from the fact that each unit area factor can be regarded as for a subject standing at center of that area with respect to a panel of unit area located in the square containing *P*.

Tabulations similar to that of Fig. 11 can be prepared for as many subject positions as are of interest. The method is, in every case, simple and straight-forward, but the numerical work is tedious and the procedure time-consuming. Ordinarily, a detailed analysis of this kind need not be carried out for conventional systems which are amenable to the generalized treatment developed in a subsequent section of this paper, but the method does find practical usefulness in the design of spot heating or cooling systems for which direct radiant energy exchange between the panel and the subject in a fixed position is of controlling importance.

*Generalized Distribution Patterns.* Of far greater practical significance than for calculation of absolute shape factors is the use of the basic experimental data as a means of investigating the general effect of panel area distribution on the uniformity of direct radiant exchange between the panel and a subject moving about in the enclosure. The curves showing variation of basic factor with location of the subject bring out the fact that radiating surface more than 10 or 15 ft from the subject contributes very little to the magnitude of irradiation which is experienced. From this it follows that the variation in primary exchange which occurs as an occupant moves around in a panel heated room is practically independent of room size (beyond a minimum value) but is governed by the varying pattern of that part of the total panel which is within a 10 or 15 ft radius from the occupant's instantaneous position.

Even if the entire ceiling is heated this pattern will still be subject to change since for a centrally located occupant direct radiation is received from all sides, but for an occupant in a position near one of the walls the 15 ft radius pattern gives no heating from one side and much from the other. The basis of the investigation of optimum generalized patterns thus reduces to a study of the manner in which the shape factor of the occupant varies with respect to that part of the total panel area which, for any occupant position, lies within the critical radius. Thus the shape and size of the room (beyond a minimum size, to be established for each ceiling height) are only of secondary importance and the major interest centers on variation of direct irradiation experienced by an occupant as he takes up successive positions along perpendicular center lines in a generalized room of any size and shape.

Further simplification of the problem is attained by shrinking the panel width down to a line (the panel center line) and investigating the effect on uniformity of a variation in the center line pattern only. By this artifice generalized results are obtained which are exact only for real panels of small width, but which lead to conclusions of adequate engineering accuracy for use with actual panels of any necessary width. The entire study can then be conducted in terms of center lines; the final generalized pattern is applied to a particular room to obtain the lineal feet of optimum panel and this figure then divided into the requisite panel area (determined by solution of the heat balance equations) to determine the necessary width of panel for that room.

Before proceeding to the search for generalized patterns a brief investigation of the relationship between point and line panels and the influence of panel length on variation in uniformity along lines normal to, and parallel to, the line panel is in order. From such an investigation further simplifications in procedure will become evident. Fig. 12 compares the shape factor variation experienced by a subject moving along a 10 ft center line when irradiated by means of either one or two point panels (panels of small area,

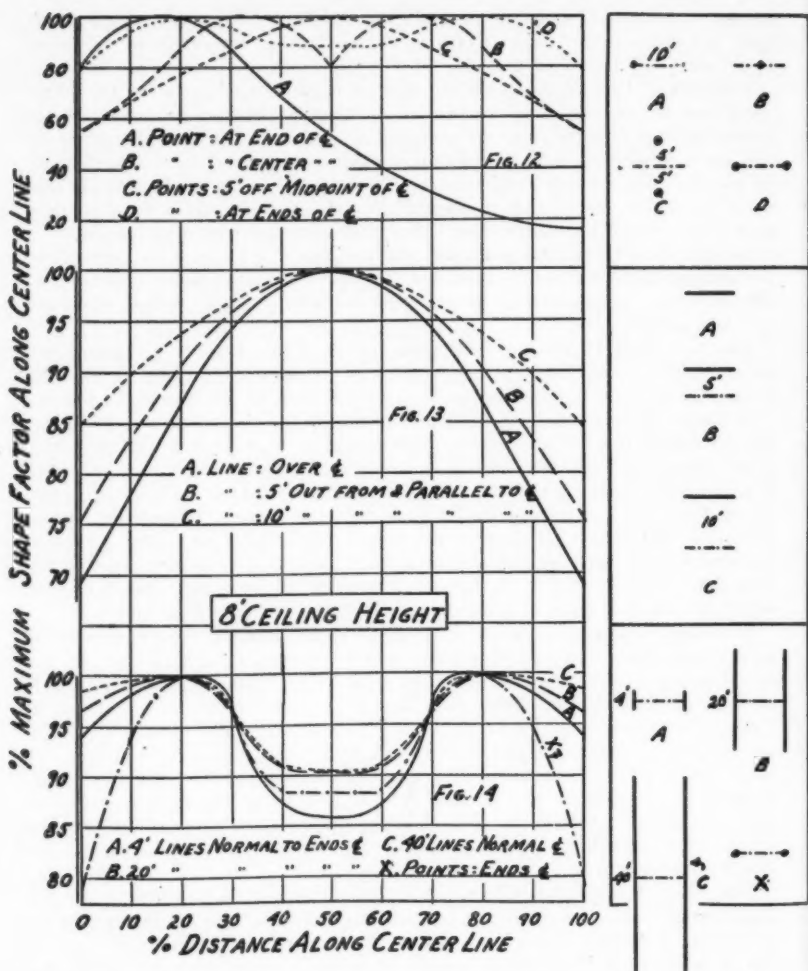
as one square foot) located at the ends, the middle, or to one side of the center line; the data are for an 8 ft ceiling and can be verified either by reference to the curve of Fig. 2 or the table of Fig. 11.

Curve A of Fig. 12 shows the per cent of maximum shape factor for points along the center line when irradiation is from a single point panel located in the ceiling directly over the left end of the center line. The shape of this curve is identical with that obtained experimentally and the non-uniformity of heating effect—from 16 per cent to 100 per cent—is obviously very serious. Curve B shows the variation when the point panel is moved to a position at the middle of the center line. For this case the maximum variation is from 55 per cent to 100 per cent and for the 6 ft section of center line terminating 2 ft in from either end the maximum variation is reduced to 20 per cent (varying from 80 per cent to 100 per cent of the maximum absolute value).

Curves C and D are for twin point panels. In Curve C the points are located 5 ft to right and left of the center line on a line normal to its mid-point; the maximum variation is 44 per cent, just as for Curve B, but the distribution is now changed so that the maximum occurs at the mid-point of the center line. This type of distribution is characteristic of off-center point panels and will be shown also to closely represent the typical distribution for line panels parallel to center line. Curve D gives the least variation, 20 per cent, of any point grouping shown in the figure. Comparing Curves B and D shows that location of the points away from the center serves to remove a large part of the drooping end characteristic which appears for B and C; further study will show that there is an optimum point spacing at which the end droop is a minimum for each particular ceiling height.

By adding consecutive point shape factors and dividing by the number of points considered, line shape factors can be obtained. Fig. 13 shows the variation along a 10 ft center line due to 10 ft line panels located in an 8 ft ceiling, parallel to the center line, but at varying distances out from it. Curves A, B, and C are, respectively, for a single line panel located 0 ft, 5 ft, and 10 ft off center. Note that the maximum variation is reduced from 31 to 25 per cent to 15 per cent as the panel is moved out—this characteristic is typical of off-center parallel panels and is helpful to the designer since it means that the selection of a symmetrical pattern giving good center line uniformity will usually give even better uniformity along any other parallel line to one side of the axis of symmetry.

Fig. 14, for an 8 ft. ceiling, illustrates the effect on uniformity of line panel length for a design having parallel line panels normal to, and with mid-points on, the ends of a center line of fixed length. Curves A, B, and C, for 4 ft, 20 ft, and 40 ft line panels, respectively, show that the center line variation markedly improves (from 19 per cent to 9 per cent variation) as the line panel progressively increases in length. This means that a distribution investigation based on a conveniently short (as 10 ft) line panel will give conservative results when extended to a system involving greater length. For comparison Curve X has been added to show the variation for lines of zero length (point panels); it is noteworthy that the general shape of the distribution curve for end-spaced points is very similar to that for lines, the principal difference being that the end droops associated with points rapidly smooth out as line panels of increasing length are used. This close resemblance between line and point characteristics is of primary importance since it



indicates a possibility of analyzing unknown panel patterns in terms of the simpler point analysis, then extending the results, conservatively, to the actual case of equivalent line panels.

**Generalized Pattern for 8 Ft Ceiling.** Figs. 15, 16, and 17 present the complete analysis leading to a determination of optimum pattern for ceiling panels in a room with 8-ft ceiling height. All data used in obtaining points for plotting the curves were obtained from the semi-profile curve of Fig. 2:

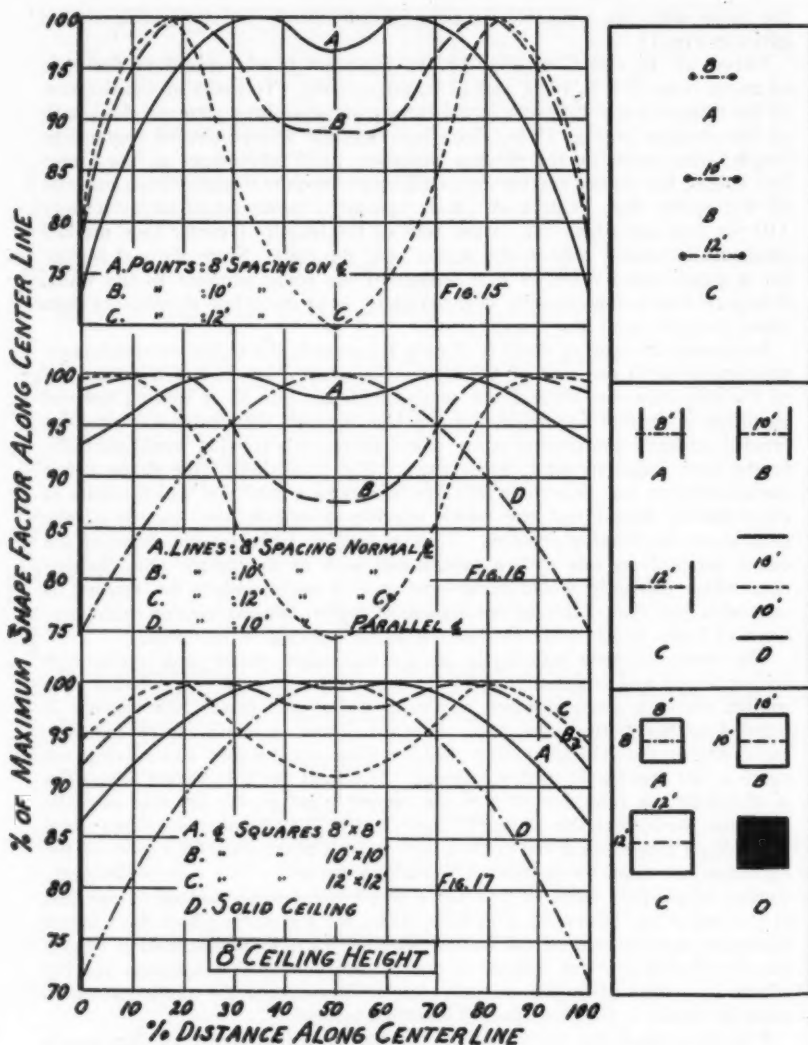
the same data are available in more useful form (for this problem) as given in Fig. 11.

Curves A, B, and C of Fig. 15 are for point panels spaced at the ends of center lines of 8 ft, 10 ft, and 12 ft, respectively. To assist in visualization of the comparative differences in uniformity of these three patterns, the length of the abscissa in Fig. 15 has been held constant irrespective of center line length; thus points on the abscissa represent fixed percentages of the center line length, but do not correspond to the same absolute distance from one end of the center line. Curve A (8 ft spacing) shows excellent uniformity (10 per cent variation) for 75 per cent of the length of center line, marked departure occurring only in the region near the ends. Since the end region, for a panel-heated room, is that section of the room adjacent to the walls, it appears that non-uniformity in this vicinity is of much less significance than when it occurs near the center.

Increasing the spacing to 10 ft (Curve B) extends the region of satisfactory uniformity to 90 per cent of the center line length, but causes a falling off of the exchange rate through the central section. For 12 ft spacing the end condition is further improved, but the loss through the center has now increased seriously and unsatisfactory distribution exists in what would normally be the most important part of the room. The transition of the shape factor variation curve for these three spacings illustrates a condition which occurs at every ceiling height and one which provides a clue to the solution of the generalized distribution problem. Thus as spacing increases the shape of the curve varies from one with a pronounced peak at the center (the limiting case which evidently would occur for an 8-ft ceiling when the spacing is somewhat less than 7 ft) to one having a central sagging section with symmetrical peaks which move farther out as the spacing is increased.

The critical spacing responsible for the maximum central peak varies with ceiling height and is determined by the rate of increase of shape factor as a subject moves a short distance out from under unit panel. As spacing is reduced to the critical, the twin peaks of Curve A move closer together, coalescing at the critical spacing then crossing one another and moving out again as the spacing is further reduced. The limit for this second separation is obviously the condition of a single central point panel; for this case the variation would take the form of Curve B, Fig. 12. Separation of the panel to spacings greater than the critical, results in a continuous falling off of the exchange rate near the center. The limiting curve for this case is obviously similar in one-half form to that for a single point panel located at one end of the center line (Curve A, Fig. 12). Thus, for a room in which the smallest dimension exceeds that of the critical spacing for a particular ceiling height, the distribution problem reduces to one of determining that optimum spacing which is greater than the critical; for smaller rooms the optimum which must be sought is that for spacing less than critical.

Fig. 16 extends the point analysis of the preceding figure to line panels 10 ft in length. The principal difference in shape between the curves of Figs. 15 and 16 is the evident improvement in end characteristic which results from use of the line panels. With such panels the maximum variation is reduced to  $12\frac{1}{2}$  per cent for 10 ft spacing and 7 per cent for 8 ft spacing. Curve D of Fig. 16 gives the variation along a center line parallel to and midway between the line panels spaced 10 ft apart. This system presents a marked central peak which suggests that combinations of equally spaced



pairs of 10 ft spaced orthogonal line panels would give supplementary variation curves which would superimpose to establish a reasonably uniform exchange rate throughout the heated space. The pronounced peak of Curve D, clearly indicates that, for two dimensional uniformity, the variation along the other

orthogonal center line must have a sagging characteristic near the center. This requirement immediately suggests (comparing Curves A and B of Fig. 16) that 10 ft spacing would be more advantageous than 8 ft spacing. Such an assumption can be verified by plotting shape factor variation for the orthogonal systems.

Fig. 17 presents center line variation for squared panel patterns using orthogonal line panels at various spacings. Curves A, B, and C, for 8 ft, 10 ft, and 12 ft spacings, show the same trend in variation which was noted with point spacing in Fig. 15, but the effect of the line panels parallel to the center line makes itself evident in the reduced central sag (over that of the corresponding Curves in Fig. 16) and the more pronounced end sag. For purposes of comparison curve D is included; this curve is for a ceiling

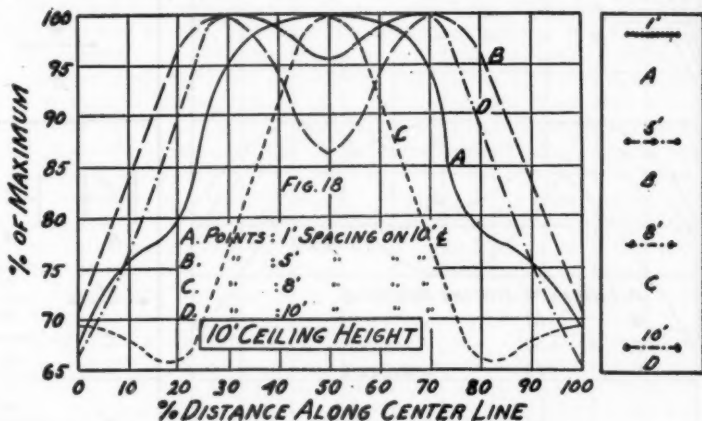
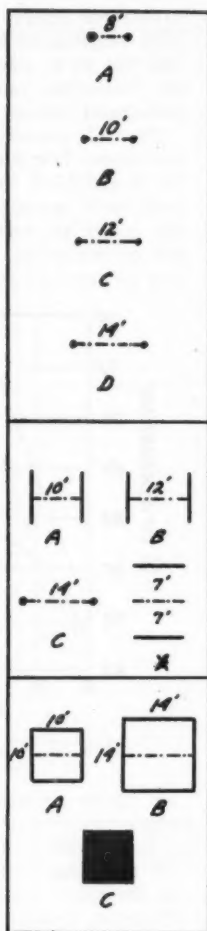
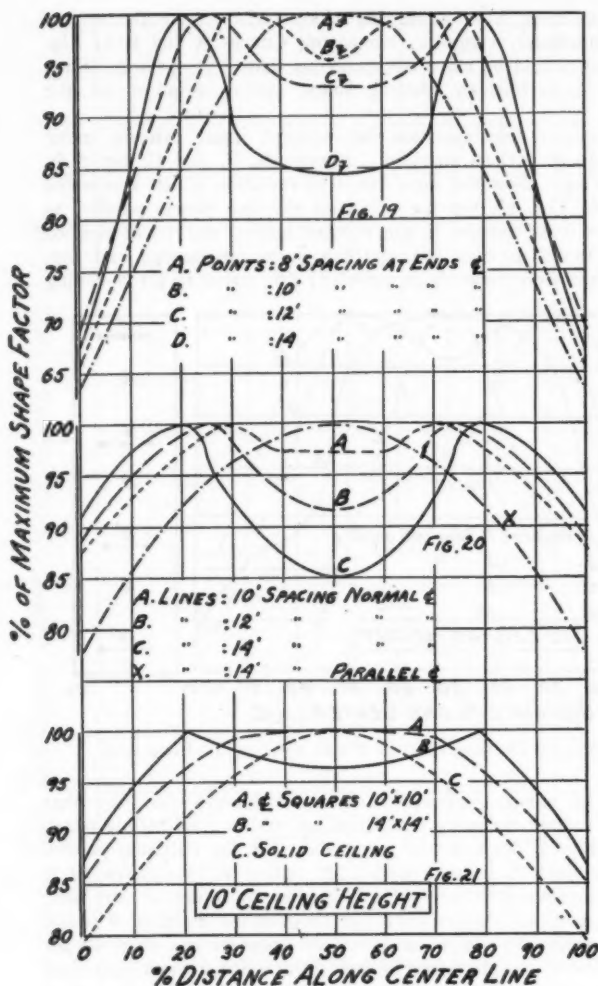


FIG. 18. CHANGE OF DISTRIBUTION AS POINT SPACING IS INCREASED

the entire surface of which is heated and it emphasizes the fact that uniformity of direct radiant exchange is not realized with a solid ceiling panel.

Consideration of Fig. 17 leads to the conclusion that the optimum pattern for the panel center lines in a room with 8 ft ceiling is in squares 10 ft on the side. This conclusion is general and applies to rooms of any size. However, squared spacing on centers anywhere between the limits of 9 ft and 14 ft will give uniformity having a maximum variation of less than 10 per cent and any such spacing can therefore be used in practice without hazard of creating regions in which the primary heating effect will be seriously out of line. Thus, for a room 9 ft x 27 ft, ceiling panels could be effectively installed as a border around the edge of the room and with two cross panels of the same width as the border, thereby establishing—with center lines—three 9 ft x 9 ft squares. There will be, of course, an alteration of the basic square variation as shown in Fig. 17, due to outlying sections of panel, but this effect, for squares greater than 9 ft on the side, can be readily shown to cause no serious alteration of the basic curves.

*Generalized Pattern for 10 Ft Ceiling Height.* The procedure for determination of optimum panel distribution for a 10 ft ceiling height is essentially



FIGS. 19, 20, 21. DETERMINATION OF OPTIMUM PATTERN FOR CEILING PANELS IN A ROOM WITH 10 FT CEILING

the same and results are shown in Figs. 19, 20, and 21. Fig. 18 shows, in Curves A, B, C, and D, respectively, the change in distribution as point spacing on a 10 ft center line is increased from 1 ft (equivalent to a line panel) to 5 ft, 8 ft, and finally to points located at ends of the 10 ft line; the critical spacing for this ceiling height evidently occurs at 8 ft since a sharp

single peak is then found and twin peaks result due to a shifting of the point panels in either direction (to 5 ft or to 10 ft) from this critical value.

Fig. 19 shows the effect of altering the spacing of point panels from 8 ft to 10 ft, 12 ft and 14 ft. From the discussion in section C it would appear that the end droop could be ignored in evaluating the relative merits of the spacings shown and that the significant region of the curves is that near midpoint of the center line. The 8 ft spacing is obviously unsatisfactory and consideration of the peak expected from the orthogonal panels which will lie parallel to the center line suggests that a spacing giving a moderate central sag as for 12 ft and possibly 14 ft, would be most likely to result in optimum squared spacing.

Fig. 20 gives the line panel pattern for 10 ft, 12 ft, and 14 ft spacing with an added curve, X, to show the distribution for 14 ft line panels parallel to and 7 ft on either side of the center line. The peak of Curve X indicates clearly that 14 ft spacing for a squared pattern, is probably close to the optimum. This is substantiated by the curves of Fig. 21 which show that the extended peaks, and slightly sagging central section characteristics, of 14 ft squared spacing gives a deviation of less than 5 per cent for over 70 per cent of the center line. Ten-foot spacing is more uniform near the center, but falls off more rapidly at the edges, while a solidly heated ceiling gives poor distribution.

The conclusion for this ceiling height is that a 14 ft spaced square center line panel distribution will give best possible uniformity of direct heating effect. Squared patterns with spacings in the range from 10 ft to 15 ft will provide a degree of uniformity adequate for most installations, but use of a solid ceiling panel would seriously affect the uniformity of exchange for average size rooms.

*Generalized Pattern for 12 Ft Ceiling Height.* Fig. 22 groups curves showing the effect of point spacing for a 12 ft ceiling height and a 10 ft center line. The curves for this case are of special interest because the critical spacing occurs at a distance, 10 ft, which frequently is significant as a room dimension. Thus, 10-ft spacing should not be used in a room 12 ft in height and the problem for this case requires investigation of spacings on both sides of the critical value. Fortunately, as is evident in Fig. 22, the optimum spacing short of the critical occurs at a value (2 ft) so close to that for a line panel that one would expect a solidly heated ceiling to give satisfactory uniformity when used in rooms having a major dimension less than about 14 ft. This case is unusual in that it is the only one, of those investigated, for which a solid ceiling gives satisfactory uniformity of direct radiant exchange.

Figs. 23, 24, and 25 give the variation for point, line, and squared center line distribution of panels in a space with 12 ft ceiling. Fig. 23 shows that surprisingly good uniformity is realized when 18 ft spacing is used and that the customary center section sag effect evidently does not commence until the spacing reaches approximately 20 ft; for 22 ft spacing the sag is sufficient to indicate serious departure (16 per cent) from uniformity of direct radiant exchange. Consideration of these curves in terms of the foregoing discussions, leads to the conclusion that a squared line panel pattern with spacing of 20 ft (acceptable range from 17 ft to 21 ft) will give excellent energy distribution in any room having a 12 ft ceiling; likewise, the same considerations indicate that, for this ceiling height, spacings between 8 ft and 15 ft should be avoided.

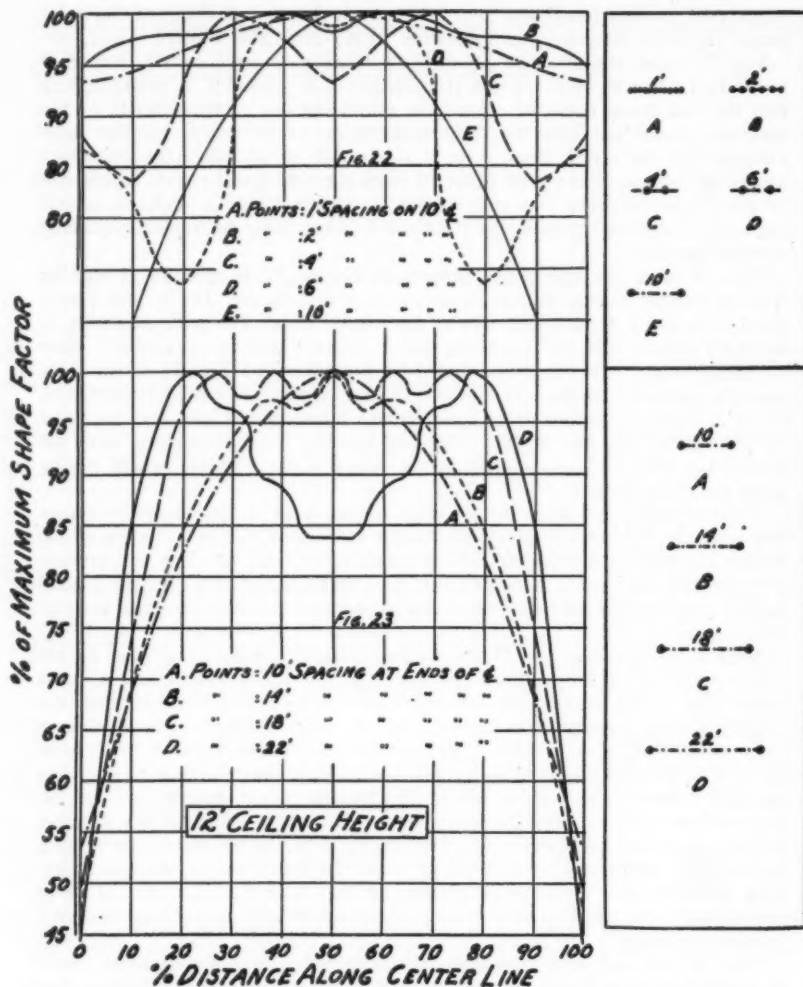


FIG. 22. EFFECT OF POINT SPACING FOR A 12 FT CEILING HEIGHT AND A 10 FT CENTER LINE

FIG. 23. VARIATIONS WITH POINT SPACINGS ON 12 FT CEILING

When the room to be heated is less than 14 ft in length or width, some pattern other than that recommended previously must be resorted to. To assist in solving this problem, Figs. 24 and 25 have been constructed. Fig. 24 shows variation along a 10 ft center line resulting from 2 ft, 5 ft, and 10 ft spacing of 10 ft line panels and from a similar line panel either normal to the center line at its mid-point or parallel to it and directly overhead. The

good distribution resulting from 2 ft spacing suggests that a solidly heated ceiling would provide acceptable uniformity for a room less than 14 ft on a side while the compensating non-uniformities of the centrally located 10 ft line panels indicate the possibility of obtaining satisfactory uniformity by installing the panels as axial strips forming a cross with center at the mid-point of the room.

Both of the arrangements discussed herein are represented in Fig. 25. From this figure it appears that the cross pattern is the more effective and

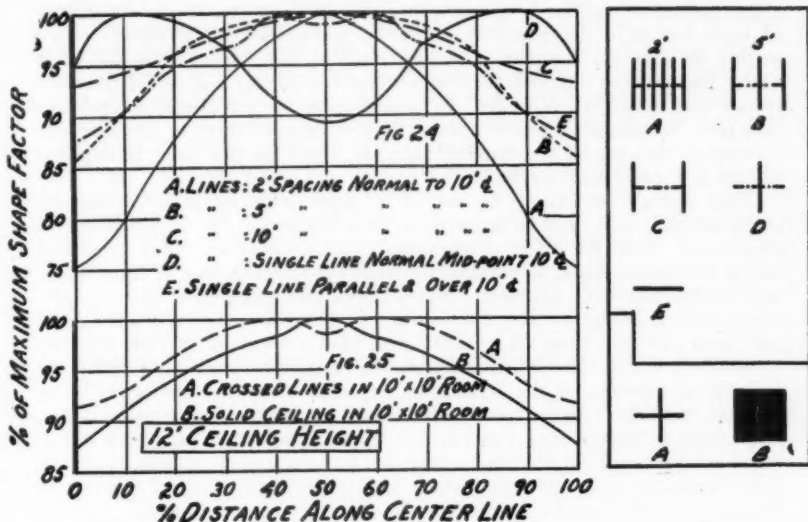


FIG. 24. VARIATIONS WITH CENTER LINE SPACINGS

FIG. 25. VARIATIONS WITH CROSSED CENTER LINES AND SOLID CEILING PANEL

should be used in preference to a solidly heated ceiling or to a solid panel located at the center of the room. However, the uniformity realized with a completely heated ceiling is satisfactory (13 per cent variation) and can be used, when heating load makes such a large area desirable, without unsatisfactory effects for rooms of any size.

The general conclusions for rooms of 12 ft height are, therefore, that complete coverage of the ceiling is acceptable for rooms of all sizes but that a square center line pattern with 20 ft spacing is the preferred arrangement for large rooms while crossed panels on the two room axes give best uniformity for small (as 10 ft x 10 ft) rooms.

**Limitations and Generalized Distribution Patterns.** The primary advantage of the generalized distribution patterns, determined in preceding sections, is in the saving in time obtained by use of such standard panel arrangements compared to that needed to evaluate the variation in uniformity of heating effect of arbitrarily selected patterns for each room considered. Generalized patterns provide a simple and convenient method of laying out

center lines for a panel distribution for which satisfactory uniformity is assured; from this basic starting point the designer will in many cases alter the pattern to meet special room conditions, yet not allow the over-all arrangement to depart so widely from the fundamental pattern as to risk serious non-uniformity.

As with any other generalized solution of an engineering problem, the assumptions and special conditions underlying the standard patterns must be kept in mind and corrections or alterations applied for rooms in which unusual factors, or wide variability of the common factors, appear. Thus, use of highly reflective wall or floor surfaces (reflective, that is, to radiant energy of wave length in the infra-red section of spectroradiometric curve) would greatly increase the fraction of energy reaching the subject from the panel (after one or more intermediate wall or floor reflections) and would therefore reduce and possibly obliterate the significance of variations in primary transfer. Fortunately, however, interreflections usually serve to increase uniformity so that use of the standard patterns would, in this case, be expected to lead to a conservative result.

Other special conditions, as use of a panel with spectral transmission characteristics, may lead to a complete invalidation of the variation curves constructed from the basic experimental data. Likewise, other conditions within the heated space, as extreme localized exposures on one or more walls, may lead to secondary exchange characteristics which can be of much greater significance than the primary interchange by direct transfer between subject and panel; in such cases an effective distribution of panel surface can only be obtained if consideration is given to the relative magnitude of direct and indirect components of the irradiation experienced by the subject. As an example, consider a square room with inside partitions on three sides and a large window located in an exposed wall on the fourth side. For such a case some compensation is obviously required for the direct radiant heat loss from occupant to the low temperature window and wall surface; this can be obtained by reducing the average width of panel along all center lines of the pattern except that one which is located parallel to and in the vicinity of the exposed wall. The amount of panel surface to be shifted to a center line near the wall, for this case, will depend on radiant loss from subject to window and to wall; thus a careful analysis would require study of the room uniformity characteristics for the individual case. In practice such individual attention to the exigencies of each particular room can not be justified, but a reasonably accurate approximation can usually be obtained by using judgment and experience as guides in estimating the extent of distortion of the standard pattern which will be required to compensate for the room's peculiarity. Note that in all such cases the effect of localized secondary heating or cooling surfaces influences only the distribution and not the total area of panel.

Deviations from the uniformity curves for standard patterns will also occur whenever the panel width is large with respect to the center line spacing of a squared pattern. This situation must inevitably arise whenever the heat transfer characteristics of the room are such that a large fraction of the ceiling area must be heated. No correction for this condition is possible (unless a higher temperature panel can be used as a means of reducing total required area), but the standard patterns are still of value since they give an exchange variation at least as good as any which can be obtained within the limitation of required total surface area.

Inadequacy of generalized patterns for use in rooms subject to special occupancy conditions: In a hospital room, for example, the heating surface could effectively be located to provide the desired effect at the location of the bed and without respect to uniformity along the axes of the room. Likewise, in factory rooms where occupants retain fixed positions, or in theaters, classrooms, and auditoriums, where the design is based on a seated subject, the generalized distribution systems developed in this paper (for a standing subject) will either be invalid or in need of adjustment. In all such cases, optimum distribution arrangements can be obtained by application and analysis of the basic experimental data presented in Figs. 2 through 10. By the same method used in obtaining the generalized pattern, for a standing figure, similar optimum arrangements can be deduced for a seated figure. Examination of the basic data for wall panels will also show that some of it can be extracted and re-tabulated or re-plotted to give the shape factors of a prone subject with respect to points on ceiling panels.

The most common reason for departing from optimum patterns in rooms where these do apply is because of the saving in first cost which is frequently to be realized by use of a simpler distribution. Study of the increased percentage deviation resulting from departure from standard patterns will show that the saving is in many cases obtainable without serious sacrifice in uniformity. In all such cases generalized pattern can be effectively used as a guide for estimating the best method of adjusting panel shape to obtain minimum loss of uniformity, yet retain an economically feasible simplicity in arrangement of both the panels and the connecting energy source (as piping, wiring, etc.).

#### SUMMARY

Basic experimental data has been presented to permit direct determination of fraction of energy received by a seated or standing subject from a wall or ceiling panel of elementary area. From those data information is also obtainable as to the fraction of energy received by a prone subject from an elementary ceiling panel. All experimental determinations are in groups of three based on the position of the subject with respect to the source both as to vertical and horizontal components and for full face, semi-profile and profile attitudes.

Use of the basic data to compute the absolute shape factor of a subject in any position with respect to a panel of any shape and size is described and illustrated; for panels of usual size in rooms having a ceiling height between 8 ft and 12 ft the order of magnitude of the shape factor is 1-2 per cent; that is, less than two per cent of the energy radiated from a conventional panel passes directly to the occupant.

Generalized patterns for the center lines of ceiling panels have been determined for the standing subject in semi-profile position. The recommended patterns are:

1. For 8 ft ceiling height use a square pattern with center lines spaced at 10 ft for best results; wherever possible spacings outside of the range from 9 ft to 14 ft should be avoided; the use of a single block panel in the center of the room is not recommended.
2. For 10 ft ceiling height a square pattern similar to that described above is recommended, but with 14 ft spacing as the optimum and 10 ft to 15 ft as the limiting range.
3. For 12 ft ceiling a square pattern with 20 ft optimum spacing is recommended (range 17 ft to 21 ft). For rooms smaller than 14 ft x 14 ft a pattern consist-

ing of centrally located crossing panels running along both axes of the room is satisfactory. Unlike rooms having 8 ft or 10 ft ceilings, a 12 ft ceiling room does have satisfactory uniformity of direct radiant exchange when the entire ceiling is used as a panel.

Limitations of the generalized patterns are discussed and the suggestion offered that the proposed standard panel distributions be used as an initial design arrangement from which to depart in seeking greater simplification, adjustment for localized exposure, or corrections for such diverse effects as inter-reflections, non-diffuse emission and the many other factors which preclude use of any completely generalized solution for optimum design of all installations.

Fundamentally, the purpose of this paper has been to present the basic experimental data which have been obtained in completion of one part of the research program being conducted at the University of California in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The analytical sections of the report and the recommendations concerning use of standard panel arrangements are intended primarily as tentative suggestions pending more complete analysis of the basic data. In all probability more effective and possibly less complex patterns can be developed; with the necessary experimental data now complete, search for such improvements can be readily carried on by anyone who cares to devote the necessary time to the rather tedious task of analyzing the data.

#### ACKNOWLEDGMENTS

The greater part of the laboratory work on this unit of the panel research project was performed by Messrs. Harold Thomas, E. H. Attix, W. R. Wykoff, Philip Ashworth, and Charles Dischler.

#### APPENDIX

The Stefan-Boltzman law gives the rate of radiant energy emission from a surface of uniform temperature and emissivity as

$$dq/dA = \sigma \epsilon T^4 \quad (1)$$

where:  $dq/dA$  = Btu/(hour) (square foot of emitting surface)

$\sigma$  = Stefan-Boltzman constant

$$= 0.172 \times 10^{-8}$$

$\epsilon$  = emissivity of the emitter

$T$  = temperature, degree Fahrenheit absolute.

For a perfectly diffuse source the rate of emission in a given direction, per unit area projected on a plane normal to that direction, is independent of the direction (as measured by angle  $\phi$  from a normal to the emitting surface) and is related to the total emission by the equation,

$$dq/dA = I\pi$$

The irradiation rate at an infinitesimal surface  $dA_2$  due to energy emitted by surface  $dA_1$  is then

$$(dq/dA_1)_{1 \rightarrow 2} = I\pi (\cos \phi_1 dA_1) \left( \frac{\cos \phi_2 dA_2}{r^2} \right)$$

where  $(\cos \phi_1 dA_1)$  = projected area of  $dA_1$  on plane normal to direction of emission.

$\left( \frac{\cos \phi_2 dA_2}{r^2} \right)$  = element of solid angle through which energy leaving  $dA_1$  passes to  $dA_2$ .

$r$  = distance between mid-points of  $dA_1$  and  $dA_2$ .

$\phi_1, \phi_2$  = angles between direction of  $r$  and respective normals to surfaces  $dA_1$  and  $dA_2$ .

Re-writing the above equation in more convenient form and integrating both over surfaces,

$$(dq/dA)_{1 \rightarrow 2} = \sigma e_1 T_1^4 A_1 \left[ \frac{1}{\pi} \int A_1 \int A_2 \left( \frac{\cos \theta_1 \cos \theta_2 dA_1 dA_2}{r^2} \right) \right] \dots \dots \dots (2)$$

Comparison of Equations 1 and 2 shows that they differ by the bracketed term of 2. This term is a function only of the geometry of the system and is therefore called the shape factor; thus, the shape factor is numerically equal to the fraction of energy striking surface  $A_2$  of that emitted by  $A_1$ .

## DISCUSSION

L. T. WRIGHT, JR., Ithaca, N. Y.: First I would like to compliment the authors on making available some extremely important data in the problem of radiant heating. Did the authors determine the net area of the human body which was used as the external area exposed to radiation?

In the case of a full-face view, is it meant that the body was turned so that a normal through the center of the body would pass through a normal from the ceiling panel?

I would like to disagree to a certain extent with the authors' last statement that the heat transfer characteristics do not enter into the distribution problem. As I see it, they do, because in a room it is not a matter of just getting the radiation uniform over the entire room—that is the irradiation from the ceiling panel—but it is a matter of getting equal comfort conditions throughout the room.

If there are two exposed walls of very poor thermal construction, the radiation from the human body to those walls would be rather large, on a severe day. When the subject is near the interior walls of the room the amount of radiation from the body to those walls would be considerably less. In my opinion, the entire room and the heat transfer through the various surfaces, have to be considered in order to get the optimum distribution of the heating surface.

R. A. MILLER, Pittsburgh, Pa.: In this panel heating I believe we are dealing mostly with flat panels. I wonder whether anything has been done also to consider the effect of the geometrical shape of the panel itself—whether it be convex or concave; whether any effort was made to concentrate or distribute the radiation from that heating panel.

PROFESSOR HUTCHINSON: Mr. Wright's third point is a good one and his disagreement is well founded. Obviously, if you are working in a room in which there is a very cold exterior surface, such as a glass area, the center line of the panel will have to be displaced. I will therefore qualify my previous statement by saying that the basic distribution pattern is determined for a room in which there is practically uniform surface temperatures of walls and floor except for such variation as may occur due to the panel itself. In any other type of room, such as one with local cold exterior surfaces, the surfaces are obviously acting as cooling panels and consequently one of two things must be done: (1) Correct the initial distribution by an analysis such as the paper presents. (2) Investigate the distribution of the local cooling surface and superimpose its effect upon the necessary distribution of the heating panel surface as determined for the heating panel alone.

In his first question Mr. Wright asked about determination of net body area. We were not concerned with body area as we used a mechanical integrator to determine the shape factor experimentally. However, our dummy had a convective loss area of 19.5 sq ft and 16 sq ft effective radiation surface.

Our definition of *full face* is in agreement with that stated by Mr. Wright.

Mr. Miller's question about the shape of the panel is extremely interesting. We have not investigated this subject, but such work is being done by others and, following the war, it will undoubtedly be published. In passing, it can be said that a great

deal of progress is being made on the use of parabolic reflectors (we mentioned them in this paper) as means of attaining local heating. For example, in a room such as this, if an individual were working in a corner and it were uneconomical to heat the entire room, it would be possible (by means of parabolic reflectors) to use a small source and with it provide sufficient local radiant heating to maintain comfort; this is being done in marine and aircraft work at the present time.

F. E. GIESECKE, College Station, Tex.: The authors have done splendid work at the University of California and their paper should be discussed fully. I wish to say a few words, hoping that it will lead to further discussion.

As I see it, it would be wise, in our practical work, to distinguish between panel heating and radiant heating. I believe there is a difference between the two. If I sit in front of a fireplace with a good fire I feel the radiant heat, and this I would call radiant heating. I was at Purdue University recently and found Professor Miller in his laboratory, which has many large windows and was not well heated. Professor Miller was seated at his desk with an electrically heated screen directly behind him and only about 2 ft away. I could call that radiant heating.

In a room, like the one in which we are, the air near the ceiling is warmer than the air near the floor. The ceiling is heated by the air and thereby becomes a heating panel for the room. So the heating of this room might be classified, in part at least, as panel heating.

At the University of Illinois the *Institute of Boiler and Radiator Manufacturers* has a research home in which the boiler is in the basement, and the heating mains are exposed, consequently the basement is fairly warm. It was found that the temperature of the lower surface of the ceiling above the basement was 72 and the temperature of the concrete floor surface about 66. Consequently, the entire ceiling was a heating panel for the basement. The entire floor was also a heating panel, because the floor had been heated by radiation from the ceiling and was warmer than the air in contact with it. As the heat from the ceiling was radiated to the floor it was also radiated to the walls. I do not know the temperature of the walls but it is evident that the ceiling, the floor, and the four walls were heating panels for the basement. In a case like this one the difference between the mean radiant temperature and the air temperature, when static conditions exist, is generally not more than 2 or 3 deg, so I believe there is no fundamental difference between panel heating and any other type of heating, but I believe there is a distinct difference between panel heating and radiant heating.

With panel heating it is well to use the entire ceiling and/or large portions of the walls, and/or the entire floor, as heating panel or panels, because the uniformity of the heat distribution in a room is increased as the size of the heating panel is increased.

PROFESSOR HUTCHINSON: I would like to add one comment, with respect to the wall-floor-ceiling controversy. I think you will find that in one of the early co-operative research progress reports\* on our work we reached certain very definite conclusions and have already expressed them in that paper.

In conclusion, for me to further discuss Professor Giesecke's remarks would be a very great impertinence because my original interest in the subject of panel heating was in large measure inspired by him, so all I can say is that I am greatly honored that he has chosen to discuss the small things that we have been attempting to do.

R. E. HACKER,<sup>4</sup> Fort Lee, N. J.: I would like to ask what is the effect when mechanical ventilation is introduced with radiant heating, such as in school classrooms.

PROFESSOR HUTCHINSON: I will answer that somewhat evasively by saying that in a paper which we presented to the Society about a year ago the subject of ventilation was covered in detail, and, since the question requires an answer that is not too simple, I suggest that you refer to the earlier paper.\* Ventilation is, of course, ex-

\* Loc. Cit. Note 1.

<sup>4</sup> Hacker & Hacker, Architects, Fort Lee, N. J.

tremely important; equations were given and a curve presented in the earlier paper showing how the effectiveness of a radiant or panel system would vary as the ventilation rate is changed.

K. C. WOODWARD, Waterbury, Conn.: I have sat through these two or three days of sessions and I do not believe I have seen another turnout compared with what we have here this afternoon. At the recommendation of President Blankin, in the early part of January, a meeting of the Connecticut Chapter was held to consider this same subject. I do not know how many there are here this afternoon who were in Connecticut on that day when the Chapter attendance doubled any that had ever been produced before, with many of those who would have liked to have been there unfortunately hospitalized. That same interest is reflected here this afternoon.

I have been a heating man all my working life and have never seen a radiant job—I'll confess; but I am thoroughly sold on the idea. Professors Raber and Hutchinson have admirably treated the mathematics of it but I have yet to find a practical answer to the engineer's question of *how do you design it?*

In the author's dissertation he has given you the scientific mathematics of panel performance but you and I know that we cannot be bothered with such lengthy, though interesting, procedure in our normally limited design time allotment. We want to know how to design simply and intelligently.

There have been several formulae given us that say "so much per square foot of tubing buried" and so on, but that is not enough. We would like to understand more clearly what really goes on *beneath* the panel surface. The fellow who wants to lay out a job is still from Missouri and wants to be shown.

What happens *beyond* the panel surface has now been pretty thoroughly covered but little of a satisfying nature leading to clearer understanding of performance *within* the panel and intelligent design of the primary (embedded carrier) heating surface has yet been presented. The Professors must know this and I feel sure could tell us so that we could proceed in the same simple conventional way as we now go to THE GUIDE for Btu performance of this or that surface. We know how to do that, and simply want to be able to do the same thing with panel heating.

R. G. VANDERWEIL,<sup>5</sup> Waterbury, Conn.: I want to continue where we stopped a few minutes ago and discuss the influence of ventilation upon the radiant properties of a radiant heating system. I think this is a very important point, which was not taken up sufficiently so far.

Back in 1937, I was interested in the study of radiant heating systems and went down to see a hospital in Rome. They called it the Mussolini Hospital and it gave treatment to a number of miserable tuberculosis cases. I entered one of the wards in a heavy winter coat. It was January, and the outside temperature was 40 or 50 deg but the ward windows were wide open. My first impression was that this room was really cool—and still the patients walked around in light hospital clothing, and apparently felt very comfortable.

The answer to the problem "how to provide a depressed room air temperature" is relatively simple: If you have, as Professor Giesecke mentioned, a closed room with a low rate of air change, the temperature difference between air and walls must be small. (It will be found that in most cases the difference between mean radiant temperature and air is anywhere between 2 and 4 deg.) However, entirely different conditions and a highly depressed air temperature result if we open the windows, or supply large amounts of cool air, as was the case in the tuberculosis hospital.

The properties of a ceiling panel are such that approximately 65 per cent of its heat output is transferred in the form of radiation. Under these circumstances we may open the windows, get rid of the warm room air and still find comfort by means of primary and secondary radiation exchange between our body and the heating panel or the wall.

The rate of ventilation is of prime importance upon the properties of the radiant

<sup>5</sup> Chase Brass & Copper Co., Inc., Waterbury, Conn.

system. If you gentlemen want to have a *true* radiant system—that is, if you want to decrease the body's radiant heat loss—you have got to get rid of the body's surplus heat by increasing its convective loss. And the one way to provide this condition is by increased ventilation with cool air.

PROFESSOR HUTCHINSON: In Aesop's Fables, a rather interesting group of tales, he unfortunately left out one which has always delighted me; that is the story about the three men who felt very ill. One went to a specialist and subsequently lived to a ripe old age because it took him that long to carry out the procedure which the specialist dictated for his eventual health. In a certain measure it is quite probable that some of you heating engineers are in that same position with respect to those of us who pretend, occasionally, to be specialists and propose treatments which are rather long.

The second man went to a non-specialist who used rule-of-thumb methods; this patient did not have many further troubles because his subsequent life was not extensive.

The third man also went to a non-specialist, but one who had discovered a sound and simple treatment; he was cured.

I can only say that our hope and belief, with respect to panel design, is that there is a very simple treatment; but our belief is that you should accept it as your problem to find it. In our research reports we are simply pointing out what the disease does, and attempting to diagnose it, but it is up to you practical people to find the cure.

As to Mr. Vanderweil's remarks, I will only add this, the Mussolini Hospital in Rome, or possibly by now the *former* hospital in Rome, was an interesting example of heating. I had the pleasure of going through it in the summer of 1938, prior to spending a little time going through the Nazi Party Headquarters in Munich, which *was* also an interesting panel heating installation. After those two inspection trips the remaining part of my summer was spent looking at the third and far more entertaining Café de la Paix installation in Paris, which does not have to open the windows, because there are none; I would recommend it as a delightful experiment station.

C.-E. A. WINSLOW, New Haven, Conn.: I cannot forbear taking just a moment to express my deep appreciation of Professor Hutchinson's paper. I have not much more to add, because Professor Giesecke said the two things that I had particularly wanted to say.

I think it is very important to remember the distinction he drew between radiant and panel heating. I do not think I agree with him that it is an absolute distinction. I have one of the little screens he mentioned behind my chair at the John B. Pierce Laboratory; but, after all, that is just a panel that is detachable from the wall. I think the real distinction is between extremely high temperature radiation (that is what we really think of as radiant heating) and heating by moderately warm surfaces. That is where I should draw the line. I think it is particularly important, however, to remember what Professor Giesecke said about the fact that panel heating does not mean in practice the exposure of the individual to an angular effect that is very much hotter than the rest of the room.

The whole value of panel heating is that all parts of the room, all the walls, the floor, and the ceiling, and the air, after a while assume almost identical temperatures, so that actually, in a properly panel-heated room, there is less inequality of exposure than there is in the ordinary convected heating system of warming a room, because the radiant heat is transformed into convection heat as it strikes the floor and the furniture and the people. The result is that in a properly panel heated room: the occupant is exposed to a very even environment, and the inequalities that are obtained from a light or from a high temperature radiant heat source are not of such quantitative importance as one might think at first.

Then I do want to say how fully I agree with Professor Hutchinson in separating these theoretical questions from the questions of practical design. The science of panel heating, as such, is what Professor Hutchinson has been dealing with. It is

concerned with obtaining certain areas of warm surface. Now, the method of warming those surfaces is an entirely separate problem. That is an engineering problem. There are dozens of ways of doing it and so far as the ultimate result is concerned it does not matter at all whether the surface panels are heated by lights, by electricity, by steam or hot water pipes in the ceiling, or by some other procedures.

The new wing at the John B. Pierce Laboratory is heated by radiant wall heating, the walls being heated by a free-standing radiator inside the wall, with a baffle above, so that circulation of warm air is obtained within that space. Our experimental rooms are now being heated by the reflection of lamps, like those drying lamps that S. G. Hibben<sup>\*</sup> demonstrated, directed against the ceiling and against the floor. We are doing that simply because we want to know exactly how much power we are putting into our quantitative work. That is not a practical procedure today. I think it may well be, some day, if proper improvements in that type of installation are made and if unit costs of electricity go down; but so far as the ultimate effect goes it does not matter in what way you heat your surface. That is a problem of ordinary engineering, not specifically of panel heating; but, for the determination of the constants, the fundamentals that are involved at the other end in what happens after the surface is heated, I think we owe a very great debt to the group at California. It has been a splendid piece of work that they have done and we all, of course, look to Professor Giesecke, the real father of this thing—and to the group in California as having done more than anyone else to bring this important procedure within the range of practical use.

MR. MILLER: The question of means of heating these panels is a particularly interesting one from my point of view and from that of the companies with which my company is competitive. I am, therefore, hopeful that you will not in any way consider that this has anything to do with product advertising, or anything of that character. It is not so intended, and is merely intended to give you a bit of information which you may not have.

It is possible to provide what might properly be called *cold heat*, which will effectively present a radiant panel for panel heating by the use of a rather newly developed, but still not too new—tempered glass product with heating elements directly on the surface, which may be made decorative—as decorative as may be desired—and will have any heating characteristics which may be specified. The product is tempered plate glass, which both we and our competitors make, and the heating panels were developed by the European glass people before we ever started to make any. We have not made too many, primarily because at the time they were first brought over to this country the interest of the country in panel heating was practically nonexistent, and it is only within the last year or 18 months that it has come to be a matter of any particular moment to the glass industry. But glass panels properly designed, properly equipped for the correct resistance in the aluminum or other metallic bands which may be placed on the glass at the time it is tempered will provide a distribution of electric energy over the ceiling or over the walls in such place as you may choose to put these panels.

That is one method whereby panel heating may be achieved, and probably the only one which I should mention.

Y. S. TOULOUKIAN, West Lafayette, Ind.: My question is addressed to Professor Hutchinson. As I gather from a hasty glance over the paper the radiant energy measurements to the body are the basis of these curves; the shape-factors that are given here. I would like to inquire whether effective temperature for comfort, which also takes into consideration air velocity and humidity and other factors, will throw these shape-factors out of order.

DR. GIESECKE: It seems to me the discussion should not be discontinued so early. I would like to point out to Mr. Vanderweil, who spoke a little while ago about the

<sup>\*</sup> Director of Applied Lighting, Westinghouse Lamp Div., Westinghouse Electric & Mfg. Co., Bloomfield, N. J. (see A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March, 1944, p. 167.)

Mussolini Hospital, that there are some open air hospitals in Switzerland, and in that case the heating panel is placed at the ceiling, directly over the head of the bed. The energy is concentrated on the patient, rather than upon the open window, Swiss physicians say that panel heating or radiant heating is more sanitary than the ordinary type of heating, because the convection currents are fewer and for that reason the dust particles, which are apt to be infected, will settle to the ground, and will not be as effective in producing or conveying disease as would otherwise be the case.

The third speaker asked concerning the means of installing these heating panels. I can give him some leads, I think. For example, if this room were to be heated by panels, or radiant heating, there might be a double ceiling here and warm air forced between the two ceilings, so that the lower ceiling was warm, say about 86 deg. Warm air could be introduced on one side of the room near the ceiling and taken out the other side, so as to blanket the underside of the ceiling. That would provide a radiant panel. Or pipes could be embedded in the ceiling and directly over the ceiling, carrying water, because steam is not usable since you cannot adjust its temperature. Of course, if electric current does not cost too much, it may be used to produce the heat; or, if a wall is to be the panel, it can be treated the same way, or it can be made hollow and warm air passed through, so that the entire wall is warm. There are any number of ways of introducing it.

L. E. SEELEY, New Haven, Conn.: This discussion reminded me of one that was held about 50 years ago by this Society, and I want to repeat it to comfort Professor Hutchinson a little bit.

If you will read either the first or the second issue of the Society's TRANSACTIONS, you will find that Professor Carpenter told the members of the Society how to determine heat losses of a building, and he pointed out that you had to take the wall into account, as well as the glass area and the volume of the room. He gave them a formula, quite a simple formula; but, the reaction of the members was something like the reaction today. They said, "Well, that is all very well; but it is pretty theoretical." Today that thing is elementary, and I think, Professor Hutchinson, that this thing will adjust itself in much the same way.

JAMES HOLT, Cambridge, Mass.: I think panel heating has some wonderful possibilities, so far as uniformity of temperature is concerned. A few years ago I had the problem of heating a swimming pool, and I think from a comfort point of view it is probably one of the most difficult jobs that we have.

After considering the problem we decided on panel heating. We installed a panel heating coil above the deck of the pool around the four sides of the pool. I thought I was going to be able to get some very good basic data from it. However, the architects fooled me, because they insisted on having a window that was 30 ft high and 100 ft long on the south side of the pool—and consequently it was necessary to place inside of that window a second partition, about 10 ft high, about 2 ft inside of the pane of glass, and supply both ventilation air and warm air to offset some of the cold drafts from the window.

I might say that the combination of the panel heating and the warm air to blanket the window has been extremely satisfactory. I have found even on days when it was zero outside men in their birthday suits could sit on a bench just inside this second sheet of glass, with perfect comfort. In fact, gentlemen, I have sat there a good many times, just to test it out, and I have found it very comfortable. I found that you could sit or stand anywhere in that room, even close to the outside walls, with perfect comfort, in your birthday suit.

CAPT. E. H. LLOYD, Washington, D. C.: I think what Mr. Woodward had in mind a little while ago is something that I, and perhaps most of the rest of us, would like to have. I think perhaps he was a little misunderstood. We can compute from THE GUIDE, and from other information available, inside surface temperatures of walls, ceilings, or anything else. We are a little lazy, though; we like to have that translated into charts or into tables that we can use.

Similarly, from the other data which is presented in the paper, if translated, preferably by a joint committee composed of the research men and perhaps some practical men who are used to working with the trade and with a field accuracy that is necessary—perhaps we can get such tables, which will be widely applied, rather than to be dependent upon a few consulting engineers to use the somewhat complex formulae to solve such radiant heating jobs as might come along.

In other words, we would like to reduce it to as simple a method of computation as is reasonably practical, if not perfectly accurate.

H. M. HART, Chicago, Ill.: I think we are getting a little far afield; but, inasmuch as we have taken that liberty to go far afield, I am going to take advantage of that and ask a couple of questions.

One question that has come to my mind to which I would like an answer, is the relative effectiveness in panel heating of copper versus steel pipe when embedded in plaster. We have found that the heat emission from a copper pipe in air was not so great as it was from steel. On the other hand, a copper pipe used in a water heater is more effective. I am inclined to think the same thing would be true if the copper were embedded in plaster.

One other point: that, in using the information in THE GUIDE on design, I wonder where that information came from on the heat output from panels, and how accurate it is.

PROFESSOR HUTCHINSON: I will start with the questions of the last speaker. As to the source of the data in THE GUIDE, that question cannot be answered by me, as I have had no connection with that Chapter in THE GUIDE.

As to the question of copper *vs.* iron, this gets back to the same story of engineering design data that has been mentioned many times before, and there I refer to the remarks of Capt. Lloyd who a moment ago said that he is a little lazy, and therefore, wishes we would do the job of simplification. We, at California, differ from him only in that we are not a "little lazy," but a lot lazier. We are very anxious, also, to establish the idea that our laziness is congenital and there is very little likelihood that we are likely to overcome it to the extent of entering the rather interesting controversy as to how to design panels. We believe it can be done. Personally, I know it can be done. But I do not propose to talk about it. I think that is something you people should do. We are still going to try and stay out of the design controversy.

Mr. Touloukian asked whether or not, if we took into account the physiology of comfort, change would occur in the position of the shape factor curves. The curves as they appear in the paper are not subject to change for any conditions except a deviation from geometry. They represent purely geometric shape factors and are based on the Stefan-Boltzmann radiation equation. Understand, then, that the basic curves to which I have already called your attention are, we believe, exact, and not subject to any need for change due to comfort factors. The way in which you use them will change, but the data, I believe, will remain exact.

Dr. Winslow is guilty of a striking degree of modesty, which seems to me to be quite unjustified. Our work at California, particularly my own attempt to learn something about this subject, has arisen from three sources: Professor Raber, at the University, who is behind all the work, Professor Giesecke, and Dr. Winslow.

PRESIDENT BLANKIN: Professor Hutchinson referred one matter back to the Society, and that is the subject of the material in THE GUIDE. I am going to ask our technical secretary, Carl H. Flink, New York, to speak for a moment on that subject.

MR. FLINK: Information found in THE GUIDE is obtained from the members of the Society. I make that statement because we have here a number of visitors who might think that THE GUIDE is compiled by some individual. Through the Guide Publication Committee—and a great many of you have worked on that in past years—we attempt to find out who are the experts on each particular subject; and then we try to obtain from them the information required for any particular chapter.

It might therefore be said that the present status of the chapter on radiant heating represents the knowledge of the Society about the subject. This knowledge is being developed, and quite rapidly, during these past few years. It is going to take a couple of years more, however, before you are going to find in THE GUIDE the kind of chapter that a lot of you fellows desire.

Every man you meet is an expert on *some* subject and he certainly knows a lot about something. But there is no man in the Society, I believe, who could write the whole Guide and do a good job on it. For that reason we are depending on discovering the fellows who are doing work on various subjects, and radiant heating is one of the most important ones on which we need help. And so the work done by Professor Hutchinson, as well as others, just as fast as it can be put into shape so that it will be more readily understood by the Society as a whole, will be published in THE GUIDE.

W. A. DANIELSON, Memphis, Tenn.: When Professor Hutchinson spoke he failed to remember some other work that was done out of California that would give an answer to the question of the relative transmission of heat from copper and steel into masonry materials. As most of you who have been to Florida know, a great part of the domestic hot water there is heated by solar heaters. When I went to Panama I figured that we could save a few of Uncle Sam's dollars by adopting that same thing down there, and so we started to study what they had done in Florida. We found nothing in the available literature, but I did visit a lot of the manufacturers and tried to get out of them some rule-of-thumb data with which we could do our designing.

I was referred to a study made out in California, and, I think, by the University and obtained a pamphlet that was very comprehensive. The problem in a solar water heater is to take the heat from the sun and transfer it into the water. Now, you can do that by direct methods or you can heat the air up and absorb the heat into the copper pipe or steel pipe and send it into your heater, or you can do it by a combination of the two.

In about a two-line thought in the study made in California I picked up the following idea. Mix cement and sand and cover the piping with that, and there will be absorption of heat from the sun and transmission through the cement slab into the water pipe. We broke up the usual single pipe into a number of parallel circuits, and obtained a very good solar water heater. I am wondering if Professor Hutchinson or anyone else is familiar with those studies made out there and could perhaps give us an answer to the question of transmission from the pipe into masonry materials, which in our case was obtained in just the reverse direction.

I could not help but be struck in this discussion by the simplicity of the problem of radiant heating; or, as a matter of fact, any heating. Keep the heat losses from the body to the normal; that is comfort. Now, the extent to which that can be done satisfactorily accomplishes good heating, and it is probably a combination of warm air and radiant heating.

I live at the present time in an ultra-modern house heated with warm air. In the mornings the thermostat becomes satisfied before the walls are heated. The room seems quite cool while we are dressing, because the walls are cold. Apparently a satisfactory heating system must have a combination of both radiant and warm air heat.

PROFESSOR HOLT: In answer to Mr. Hart's question as to whether or not you can use steel, wrought iron, copper or other materials, it depends a good deal upon how your panels are installed. In the particular case that I spoke of we had a rather serious problem because in the case of the panels in the ceiling the panels were to be used as a backing for the plaster. We were to fasten our metal lathe directly to the coils, using the coils as a support, and then to back plaster on top of the coil with an insulating material—in this case cork—above the coils. In order to prevent cracks in the plaster it was very essential that the coefficient of expansion of the coils be the same as the coefficient of expansion of the plaster.

We made quite a few studies and found that steel and wrought iron had coefficients of expansion that were almost identical with the plaster. The same was true with the coils placed in the deck of the pool. In that case the coils were cast right into the deck of the pool, and then tiles were placed on top of the concrete. Then it was essential that we have a coefficient of expansion of our pipes, exactly the same as the concrete, or very nearly the same.

Again, the coefficient of expansion of your wrought iron and steel pipes is very nearly the same as the coefficient of expansion of the concrete. I will say that in the five years that that installation has been in service no major cracks have developed, either in the deck of the pool or in the ceiling. So evidently our conclusions were very nearly correct.

W. L. FLEISHER, New York, N. Y.: As soon as we start to talk about formulae practically everyone except the professors become tongue-tied. I think radiant heating has been discussed at these meetings for the last five or six years, and it has always had something seemingly mystical and magic about it.

Of course we are approaching radiant heating more and more, with the increasing insulation of our buildings, as Dr. Giesecke and some of the other speakers have said. It does not make any difference how radiation is produced as long as the surrounding walls are sufficiently warm so that the radiation from the body to the cooler surfaces is not too great for the metabolism of the body to keep pace with it.

My purpose in talking to you on this subject is to say that this whole idea is not new. The very first heating that we have on record for residences dates back to the first century in Rome, and it continued as the general method of heating up to the Byzantine period, about 900 A.D. Now we are coming back to it, and we are making something esoteric of it. It is magical, it is mystical—and, of course, we are very prone in this organization to swallow new ideas whole, and sometimes the hook with them.

Radiant heating is not a panacea for every heating ill and everything General Danielson mentioned. The fact that there was a warm-air heating system in the bathroom and that they are first too hot, and then too cold, applies conversely in many cases in our own houses to panel heating.

I made a very careful examination of panel heating in England where the variation in the temperature was rapid and where the mass of the structure held the heat to such an extent that it took almost two days in warmer weather for them to get to the point where they could even live in their houses.

So it is not merely a question of designing a panel, because the panel must be taken in conjunction with the mass, or the density, of the structure, and if you have variations in your climate, then you must be more careful in designing your structures so that they can lose heat fairly rapidly, as well as gain, and maintain a constant condition.

The whole tendency toward better construction, the better insulated construction that is used today, makes many of our houses, no matter how they are heated, panel-heated houses, because the danger in old methods of heating was that with poor construction walls cooled so rapidly that for a long time we were cooled before we could get warm again. That is not at all true now and in a paper<sup>7</sup> presented before the Society in 1941 this case was given and there was an eager discussion of this very subject, as to whether insulated houses, in a way, were not panel-heated houses.

I make one particular suggestion, that is, that in all our discussion, we have not mentioned the possibility of bright surface or low emissivity surfaces in our construction if we are going to use panel heating, because in that particular way we could prevent the radiation of cold from the outside entering and maintain the heat or coolness just as we liked inside. We cannot discuss this question at all, so far as our modern construction goes, without taking into consideration the other factors that enter into the building.

<sup>7</sup> Operating Results of a Residence Radiant Wall Heating System, by E. J. Rodee. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 123.)

MR. WOODWARD: The statement has just been made that the co-efficients of expansion of steel and wrought iron were the same as plaster. I am assuming that cement plaster was implied. Otherwise, I would have to suggest, if I am not wrong in so doing, that you will find this statement to be incorrect.

H. F. RANDOLPH, Utica, N. Y.: Many phases of panel heating have been discussed here this afternoon, but one question of calculation has not been mentioned. I do not see any difference, myself, in the heat loss of any given structure, regardless of the type of heating system that is installed, but the chapter dealing with panel heating in THE GUIDE gives us two methods of calculation which, in my experience, when applied to a given structure, do not agree. Obviously, therefore, one of them must be wrong.

I would like to know the opinion of Professor Hutchinson and Dr. Giesecke and some of the engineers here who have calculated these systems as to which method they use.

PROFESSOR HOLT: We investigated the particular plaster that we were using and found that the coefficient of expansion of the particular plaster was the same as the wrought iron pipe that was used. I do agree that if you take a large variety of plasters you would get a difference in the coefficients of expansion, and therefore, you must be careful, if you are embedding a coil directly in plaster, to be sure that you get a plaster that has a very nearly similar coefficient of expansion. I believe that the statement is correct that concrete has the same coefficient of expansion as wrought iron pipe.

PROFESSOR HUTCHINSON: Mr. Randolph asked if I would comment on the two Guide methods of load calculation. I will refer him to a discussion, by Professor Raber and me, submitted to the Society for inclusion in the next edition of the TRANSACTIONS, a discussion on Mr. Randolph's own paper on load calculation. In this paper Mr. Randolph presented experimental data on a given house; he then applied to that house the theoretical load calculation method of THE GUIDE and also the so-called Btu method given in THE GUIDE. He found that the two answers were quite different, not only in themselves, but also from the experimental results. Professor Raber and I went through the load calculations for the same house using what I will call the Society's method, the rational method based on our research, and we found much closer agreement with the experimental data. We found that this method led to very much better agreement than did either of the two others and it seemed to indicate that the extent of difference was within the range of experimental error.

I mention this merely to again emphasize that based upon the work to date, that it is possible to take the rational procedures which have been evolved from the co-operative research and develop and simplify them for practical purposes.

As to why you should do it, Mr. Fleisher said that all except professors are practically tongue-tied when it comes to discussing equations; all I can say there is that we unfortunately lose our glibness when it comes to discussing the desired simplified calculation procedure.

As to Mr. Fleisher's remark about the necessity of our considering reflectivity, we have already done so. We presented to the Society, some time ago, a research paper on the performance of panel heating and cooling systems,<sup>8</sup> in which was included data from a comprehensive series of tests showing the way in which reflectivity affects comfort within a room.

As to the remark made by one of the speakers that panel heating is old stuff, that need be no surprise to any of you; refer only to Webster's definition of the word "hypocaust."

<sup>8</sup> Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 35.)



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## SEMI-ANNUAL MEETING, 1944

### Grand Rapids, Mich.

One of the most important actions taken by members of the American Society of Heating and Ventilating Engineers attending the Semi-Annual Meeting 1944 in Grand Rapids, Mich., was the adoption of a resolution offering "the full support of the members of the A.S.H.V.E. to the Government in the National Fuel Efficiency Program to expedite the war effort by helping to prevent the needless waste of heat and power."

The Meeting attracted 352 members, ladies and guests for the four technical sessions held at the Pantlind Hotel, Grand Rapids, Mich. This Grand Rapids Meeting was distinguished by the unusually heavy attendance at the technical meetings, and the entertainment planned for three hours was original and appealed strongly to the visiting audience.

Pres. S. H. Downs, Kalamazoo, Mich., called the first session to order and a greeting for the visiting members was given by H. D. Bratt, president of the Western Michigan Chapter.

A brief response was made by President Downs after which a series of By-Law Amendments and Research Regulations were presented by Secretary A. V. Hutchinson. The new By-Laws on rate of initiation fees and dues are as follows:

**ARTICLE B-IV—Admission Fees and Dues.** *Section 1.* The admission fee of Members and Associate Members shall be ten dollars (\$10.00); of Junior Members five dollars (\$5.00). Admission fee must accompany application for membership.

Motion for adoption was made by C. F. Boester, St. Louis, Mo., and seconded by E. C. Evans, Buffalo, N. Y.

*Section 2.* The annual dues of Members and Associate Members shall be eighteen dollars (\$18.00); of Junior Members ten dollars (\$10.00); of Student Members three dollars (\$3.00).

A motion for adoption was made by L. T. Avery, Cleveland, and seconded by E. M. Mittendorff, Chicago.

*Section 5.* Of the annual dues paid by Members and Associate Members forty per cent (40%) shall be allotted to the Research Fund and shall be immediately deposited in said Fund and shall not be used for any other purpose.

Motion for adoption was made by G. D. Winans, Detroit, Mich., and was seconded by H. B. Hedges, Philadelphia, Pa.

*Section 7.* The dues of a new member of any grade shall be pro-rated quarterly but if the amount paid is less than five dollars and fifty cents (\$5.50) such member shall not be entitled to receive the volume of Transactions or The Guide for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership. Forty per cent (40%) of all dues of Members and Associate Members shall be allotted to the Research Fund and shall be immediately deposited in said fund and shall not be used for any other purpose.

Motion for adoption was made by M. W. Bishop, Milwaukee, Wis., and seconded by John Howatt, Chicago.

**Section 8.** Junior Membership shall be limited to ten (10) years and upon election to a higher grade, such members shall upon notification of transfer, pay the annual dues of the grade to which they have been transferred.

Motion for adoption was made by L. G. Miller, East Lansing, Mich., and seconded by J. W. Miller, Lansing, Mich.

The several motions for adoption were unanimously carried.

President Downs announced that the proposed amendment on Committees, Article B-VIII would be presented by J. F. Collins, Jr., Pittsburgh. Mr. Collins said that Chapter Delegates Conferences had been held for a number of years at each Annual Meeting and last January the group decided they would like to be considered as a Society Committee and to accomplish this directed their chairman to appoint a committee of three to prepare an amendment to the By-Laws. It was not the purpose to take over the functions of any of the other committees, such as Membership, Chapter Relations, nor did they say that they wanted funds to come to each Society Meeting. The proposed amendment was sent to the Chapter Delegates, and 25 of the 32 expressed approval, and 4 did not vote.

Mr. Collins then offered a motion for the adoption of the amendment, which was seconded by D. L. Taze, Cleveland, Ohio.

**ARTICLE B-VIII—Committees.** *Section 14.* There shall be a standing committee known as the Chapter Delegates Committee. The membership of this Committee shall consist of one Delegate or Alternate from each local chapter, who shall be selected as each chapter may direct. The Chapter Delegates Committee shall meet at each Annual and Semi-Annual Meeting of the Society to discuss problems of mutual interest affecting the chapters. The Chapter Delegates shall elect a chairman to serve for a term of one year, from June to June. Meetings of the Chapter Delegates Committee may be attended by any local chapter officer, but each chapter shall have only one vote.

In the discussion John Howatt, Chicago, Ill., chairman of Constitution and By-Laws Committee, expressed the opinion that the proposed amendment should have been referred to the Council, and if the Council saw fit it would refer it to the Constitution and By-Laws Committee for preparation in proper form. He pointed out that in the present By-Laws, provision is made for 12 regular committees and that the By-Laws stipulate that the Council may appoint any other special committees to carry out any specific purposes. He suggested that there should be a definite clarification of the functions and duties of some of the existing committees before we create committees without any specific functions.

Mr. Howatt recalled that the Chapter Delegates Conference was established when he was president of the Society, but he felt that the Chapters Committee should do something more than just discuss matters.

Mr. Collins expressed the opinion that the division between the Chapter Relations and the Chapters Committee was quite clear cut, and felt that any recommendations made by the Chapters Committee to the Council would naturally be referred to the Chapter Relations Committee.

Mr. Avery stated that the discussion brought out the fact that there was a question whether the Constitution and By-Laws Committee is a steering committee, or whether it is an approval committee. He thought that it was rather unfortunate that our Constitution permitted different methods of revision for the Constitution, the By-Laws and the Rules.

President Downs pointed out that there was an apparent conflict between the proposed amendment and the Rules where the Chairman would serve from June to June, and invited discussion of this matter.

In commenting, C. F. Boester, St. Louis, Mo., said that the members of the Committee would serve from January to January, but only the chairman would serve from June to June. He believed that this was desirable because many of the chapters changed officers at their May meetings.

Motion for the adoption of *Section 14*, Article B-VIII, was carried.

On motion of Mr. Collins, seconded by Mr. Boester, it was voted that the present *Section 14* of Article B-VIII of the By-Laws, edition of 1943, shall be renumbered *Section 15*. This motion was unanimously carried.

President Downs then announced that the following amendment was submitted in accordance with the vote of members at the 50th Annual Meeting in January 1944, and was prepared by the Constitution and By-Laws Committee and endorsed by the Council:

**Addition to Article B-IX—Election of Officers.** *Section 1.* . . . . However no member of the Society is eligible for nomination for election to serve on more than one elective body of the Society at a time.

**Addition to Regulations Governing Committee on Research—Article II—Organization—Section 1 (additional item f) (f)** No member of the Society is eligible to serve on more than one elective body of the Society at a time.

It was moved by Mr. Howatt, seconded by Mr. Evans, that the amendment be approved.

President Downs put the motion to a vote and it was defeated.

A similar amendment to Research Regulations was presented and on motion of Mr. Boester, seconded by C. H. Pesterfield, it was voted to reject the amendment to Research Regulations, Article II, *Section 1f*.

President Downs introduced Paul D. Close, Chicago, Ill., who presented the first paper on Selecting Winter Design Temperatures (see p. 281).

Following a three-minute intermission, President Downs announced the next paper would be given by L. T. Wright, Jr., Ithaca, N. Y., on the subject of Periodic Heat Flow—Homogeneous Walls or Roofs, who was the co-author with C. O. Mackey (see p. 293).

The final paper in the morning session was given by E. K. Campbell, Kansas City, Mo., on the subject of A Method of Heating a Corrugated-Iron Coal Preparation Plant (see p. 313).

First Vice-Pres. C. E. A. Winslow, New Haven, Conn., presiding officer of the second session, Monday afternoon, announced that the session would be devoted to a discussion of the subject of fuel conservation. He referred to the Society's contribution toward the war effort through its promotion of the fuel conservation program during the past year and also through the 10 per cent of its membership now in military service.

K. C. Richmond,\* Chicago, Ill., introduced the subject of Fuel Conservation by giving a prepared discussion on *Education—The Engineer's Major Job in Heating and Fuel Conservation*. He pointed out that the ten basic needs if the problems of fuel conservation are to be solved were:

1. Familiarize all concerned with the importance, needs, possibilities, advantages and opportunities in saving heat, steam or power.

\* Editor, Coal-Heat, Member of A.S.H.V.E.

2. Sell more fuel and heating dealers on the value of getting in touch with every customer so as to help insure the best results from the fuel or equipment used.
3. Encourage cleaning and inspection of every possible fuel using plant.
4. Provide adequate instructions on better maintenance and operation. Give fuel users a "Heating Service Report" to show specifically what they can do.
5. Reduce all possible heat, steam, power and fuel losses.
6. Assist users of non-familiar fuels.
7. Increase publicity, training and educational efforts.
8. Encourage the use of insulation and other means of fuel savings that do not require the use of critical war materials.
9. Discourage overheating for reasons of health protection or the prevention of colds and absenteeism.
10. Set up standards of performance, comparison, costs, quantities, and results to show what can be—is being done.

A recent survey disclosed:

1. That less than 20 per cent of the heating plants are in "good condition"—20 to 30 per cent in "fair condition"—60 per cent or more generally are in "poor condition"—depending on the community.
2. In answer to the questions on "housekeeping" or the way most heating systems are operated, these men declared that less than 10 per cent of the heating plants including those in apartment houses and janitor operated buildings in general, are clean, and are skillfully operated. About 20 per cent are *fairly well* operated.
3. Unfortunately, time does not permit the use of a number of most interesting direct quotations commenting in detail on the faulty operation and maintenance of the majority of heating systems.
4. According to these authorities, 50, 60, 75, 80—even 90 per cent of these plants could be modernized or replaced to advantage as we go out of the war period. Certainly, a substantial percentage is now in need of major alterations or repairs—overhauling, modernization or replacement.

Obviously, the responsibility for the general faulty operation and lack of proper maintenance of so many heating facilities—and the needless waste of heat and fuel rests squarely on the shoulders of: (1) building owners and operators themselves; (2) fuel producers, equipment manufacturers and their distributive connections; (3) on architects, builders and contractors; (4) on labor, trade and other associations; (5) on governmental authorities.

But the facts are that most fuel users have not come to appreciate the personal importance of proper operation and maintenance of their heating facilities. Yet, the individual who signs the checks just can't evade the responsibility for overall heating costs or waste. Too often he has pinched pennies—and wasted dollars.

Nine out of ten fuel using plants need attention—inspection, cleaning, adjustment, alterations, repairs—but they are not getting such attention because the owners haven't been sold. So, "education" and selling them is the engineer's job today. It will help win the war, too.

Today—customers are tremendously interested in their heating problems—even if many haven't paid any too much attention to them in the past.

Most home or building owners will appreciate the good faith, interest, and patriotic motives of such efforts to help them improve their heating results, prevent heat losses and eliminate fuel waste.

Discussion on Mr. Richmond's address was postponed in order that it might be included in discussion of the subjects which were to be introduced by the members of the Panel on fuel conservation.

#### PANEL DISCUSSION

Following a brief recess, Chairman Winslow introduced Past President M. F. Blankin, Philadelphia, Pa., to lead the panel discussion. Mr. Blankin

introduced the following members of the Panel: R. F. Connell, Detroit, Mich.; R. C. Champlin, Detroit, Mich.; M. W. Crew, Milwaukee, Wis.; C. C. DeWitt, Lansing, Mich., and John James, Cleveland, Ohio.

Professor DeWitt, Michigan State College, representing the National Fuel Efficiency Council and Co-ordinator of the Greater Lansing Area, presented the reasons for the anticipated fuel shortage during the coming winter. He stated that the average householder probably would receive 75 to 90 per cent as much coal during the coming season as in the past. Among the reasons for this reduction he mentioned some of the following: The armed forces and war industries require 6 per cent more fuel; railroads require 10 per cent more fuel; and transportation facilities available for moving fuel would be decreased through the necessity for moving war goods. He stated that immense quantities of petroleum, gas, and coal were being diverted for Army and Navy use and to supply temporarily certain liberated areas.

The national stockpile of coal above ground was assumed to be less than 30 days ahead of solid fuel demands. The anticipated demand for bituminous coal for the current season was given as 626,000,000 tons compared to an anticipated production of 596,000,000 tons. The difference between the two indicated an anticipated shortage of 30,000,000 tons. Anthracite coal anticipated requirements were 65,000,000 tons, which was about 8,000,000 tons above the anticipated production of 57,000,000 tons. The total shortage of anthracite and bituminous was therefore given as 38,000,000 tons and it was expected that the whole nation must keep going and keep warm on a third of a ton less fuel per inhabitant than had been available during the past year.

Professor DeWitt explained the functioning of the National Fuel Efficiency Council. It was organized from the ranks of patriotic business and technical men serving without pay. The Council in cooperation with the Department of the Interior and through the agency of the Bureau of Mines had inaugurated a National Fuel Efficiency Program, which was expected to reduce fuel consumption to the quantity available. The National Fuel Efficiency Program was only slightly concerned with the average householder's use of fuel. Its field of activity included all users of fuel beginning with four apartment buildings and extending to the largest commercial user.

In organizing the National Fuel Efficiency Program each State is divided into areas centered about large industrial cities and a coordinator is appointed for that area. The supervising engineer, Thomas C. Cheasley, Kansas City, Mo., selects coordinators for the fuel area and the engineers at the Bureau of Mines, Pittsburgh, act as liaison officers between the coordinators and the supervising engineers. It is the duty of the area coordinators to organize an advisory committee which, with the approval of the coordinator, selects regional engineers who operate on a volunteer basis.

The regional engineers call upon the fuel users to enroll in the fuel saving program. The fuel user is asked to sign a pledge and this pledge eventually is sent to the National Fuel Efficiency Council in Washington.

Advice for fuel users has been prepared in the form of 75 different types of information sheets referring to equipment affecting the quantity of fuel used. The assistance of smoke commissioners and smoke abatement engineers from all parts of the country had already been promised and much assistance was expected through the wholehearted cooperation of engineering societies.

Emphasis was placed upon the voluntary nature of this fuel saving program which was sponsored and supported by industry.

Chairman Blankin thanked Professor DeWitt and expressed the conviction that the A.S.H.V.E. was enthusiastically back of the program in which many members were serving as region engineers and coordinators. He urged Society members to consult local coordinators and to offer their services.

Mr. Connell was presented as the speaker who would discuss the subject of the *Heat Generator*. The first point made by Mr. Connell was that the engineer would probably be faced with the problem of obtaining fuel savings by using the existing equipment, rather than by substituting more efficient appliances. Much could be done by repairing fire doors, smoke hoods, control equipment, etc., and eliminating air leakage. In the examination of boilers, for instance, it would be expected that in some instances incorrect assembly of sections would be discovered as a cause of wastage of fuel. Instances were cited in which incorrect sections had been used for repair purposes with the result that flue gases had been able to short-circuit some of the flues. Reference was made to the opportunity for saving fuel by insulating the boiler and piping, which in certain sections of the country were usually left uninsulated.

Mr. Connell felt that the need for fuel conservation would promote a better understanding and appreciation of the type of equipment required for efficient operation and that this interest would be mutually advantageous to the purchaser and manufacturer.

Chairman Blankin mentioned that unclean flues could cause an efficiency loss of 6 per cent. He pointed out that if half of this could be saved, it would amount to 120,000 carloads.

The next speaker was Mr. James, whose subject was *How Can the Stoker Industry Assist in Saving 38,000,000 Tons of Coal in the Coming Heating Season?* Mr. James proposed three methods in which the stoker industry should be particularly interested: (1) The conversion of hand-fired installations to stoker operation in order to overcome manpower shortages, enable the use of a local coal which might be of an inferior grade, and to reduce smoke difficulties. He pointed out that by use of automatic firing and increased plant efficiency there might also be obtained an accompanying increased steam output without the necessity for increase of boiler capacity. (2) Modernization of existing stoker installations was mentioned as a second way in which the stoker industry could assist in saving fuel. Mr. James stated that of the 1,000,000 stokers now sold, 75 per cent were of domestic or residential types and 25 per cent were of commercial or industrial sizes. (3) A general check-up of the heating plant efficiency was advocated as a third method of saving fuel.

Chairman Blankin introduced Mr. Champlin, to whom the subject, *Ways and Means of Utilizing Oil Efficiently*, had been assigned. He stated that any fuel conservation program must be based upon three main factors: (1) combustion efficiency; (2) heating plant efficiency; (3) conservation of heat loss from the building. He discussed the first two items because they were particularly applicable to the present panel discussion.

Under combustion efficiency Mr. Champlin stressed the need of determining the rate of oil supply required for a given installation and then making the air adjustments correctly to obtain a minimum flue gas loss. He mentioned the importance of a draft regulator to prevent variation in excess air. The importance of preventing leaks of air into the appliance or into the combustion chamber was explained. Prevention of soot formation and the cleaning of scale, rust, sediment, etc., from heating surfaces were mentioned as important items in improving efficiency.

Mr. Crew, chairman of the Fuel Conservation Council for War of the Automatic Control Industry, discussed *Fuel Saving as Effected by the Use of Automatic Controls*. He reported an estimate that proper control equipment could save 6,000,000 tons of coal. This estimate was based upon studies of records of the *National Warm Air Heating and Air Conditioning Association*, the *Stoker Manufacturers Association*, and the *National District Heating Association*. He mentioned that the saving could be obtained by manufacture of 375,000 damper regulators, 400,000 barometric controls, 35,000 sets of heating system controls, and 30,000 sets of commercial and industrial boiler controls. It was estimated that the production of this equipment could be obtained by adding only 735 people to those presently engaged in making controls. About 50 or 60 of the 735 people required would need to be skilled mechanics. It was expected that a saving of 1,200,000 miner man days would be obtained because of the reduction in fuel requirements resulting from the use of the proposed control equipment. Mr. Crew stated that the War Production Board had granted material for the production of the control equipment.

It was Mr. Crew's estimate that 22 per cent of existing stoker installations in the United States were fully equipped with draft control. He reported a survey of 112 stokers in one city which showed a flue gas temperature of 425 to 875 deg, with an average of about 700 deg. Mr. Crew then referred to the publicity program which would be used to make owners aware of the need for automatic controls as fuel saving devices.

Chairman Blankin announced that the meeting was then open for discussion from the floor.

E. E. Dubry, Detroit, pointed out that fuel would be saved by what happened in the boiler room after the pledge had been signed. He pointed out that firemen have definite ideas about the condition in which they prefer to keep a fire and that, in order to teach them to manipulate fires properly, it is necessary to educate them in the use of simple equipment for determining proper combustion conditions.

John Howatt, Chicago, emphasized the importance of preventing heat loss up the stack. In being responsible for operation of about 800 boilers, he had found it worthwhile to provide the operators with Orsat analysis apparatus and flue gas thermometers. He referred to the Society's participation in the oil rationing program and felt that pride could be taken in the work done.

Mr. Howatt stated that there was need for authentic information regarding the actual amount of fuel available. Statements had been made that the production of coal per miner was less than formerly due to increased age of workers, that coal would probably be needed for shipment to Europe, and that the British fleet which might be operating in the Pacific soon would require a considerable amount of U. S. coal. With decreased cutting of wood because of lack of manpower, 15,000,000 tons of coal would be required as a substitute for wood.

Mr. Howatt stated that the fuel saving program was so huge that a few hundred men would accomplish very little. The need would exist for thousands and he urged everyone capable of helping to enlist in the work.

H. E. Lewis, Toledo, pointed out that insufficient attention was being paid to efficient utilization of the heat produced. He proposed an increase in emphasis on insulation of industrial equipment. Due to the great proportion of fuel used in industrial processes, the opportunity for quantity savings would be greater and consequently he advocated increased attention to savings that could be made by large users of fuel.

Chairman Blankin asked Mr. Richmond for statistics on the relative consumption of coal by industrial and domestic users.

Mr. Richmond gave the proportion of coal used as follows: 170,000 manufacturing industries 20 per cent; railroads 25 per cent; greenhouses, laundries, dairies, apartment houses, and homes 20 per cent; coke and steel industry 35 per cent. The annual production of bituminous coal was expected to be 600,000,000 tons. In regard to production per man, he stated that in certain strip mines 16 tons of coal were produced per man per day.

Chairman Blankin called upon B. F. McLouth, Minneapolis, Minn., for some comments on the fuel conservation program conducted by the U. S. Army.

Mr. McLouth stressed the value of educating the men who were operating fuel burning equipment. He showed how proper training and comparison of results obtained in various camps had stimulated fuel saving throughout Army posts with savings of 25,000 to 50,000 tons annually in large camps.

R. K. Thulman, Washington, D. C. suggested that much fuel had been saved by the use of insulation, but that the reduction of the heating load effected by use of insulation had in some cases caused a heating plant to be too large. He felt that the insulation industry should, therefore, be interested in providing proper operating instructions for heating plants.

E. K. Campbell, Kansas City, Mo., suggested the use of economizers installed in the outlets of boilers in cases where flue gas temperatures were too high. He cited an instance in which such an economizer supplied air to heat a basement auditorium of a church.

He also suggested that proper air circulation could frequently save fuel by distributing the heat where needed and thereby eliminating overheating of some parts of the building in order to heat others adequately.

A. R. Frantz, Lansing, Mich., gave his opinion that air filters if not replaced or renewed when dirty would cause a considerable waste of fuel.

L. E. Seeley, New Haven, gave his experience in cleaning boiler heating surfaces by using a water spray applied with the burner in operation and expressed the opinion that this method of cleaning produced a very clean surface and eliminated the necessity for brushing or scraping.

Chairman Blankin thanked the members on the panel, after which Vice-Pres. C.-E. A. Winslow took charge of the session.

On motion of Mr. Blankin, seconded by Mr. Frantz, it was voted that the Resolutions Committee prepare appropriate resolutions endorsing the National Fuel Efficiency Program.

The meeting adjourned at 4:30 p.m.

The third session was called to order at 9:30 a.m. in the ballroom by 2nd Vice-Pres. A. J. Offner, New York.

W. F. Wells, Philadelphia, presented his paper on Air Disinfection in Ventilation (see p. 361).

A. D. Brandt, Chicago, presented an abstract of his paper on Industrial Exhaust Ventilation in Industrial Hygiene (see p. 331).

Chairman Offner then called upon B. H. Jennings, Evanston, Ill., who presented the paper on The Use of Glycol Vapors in Air Sterilization and the Control of Air Borne Infection, which he prepared with Edward Bigg and F. C. W. Olson (see p. 343).

Past President Blankin announced that applications for Chapter charters had been received and approved for a Rocky Mountain Chapter to be located at Denver, Colo., and a Memphis Chapter to be located in Memphis, Tenn. The charter meetings for both Chapters would be held in the early fall, he stated. Mr. Blankin also reported that the Organizing Committee at Columbus, Ohio,

had sent a telegram stating that an application for a Central Ohio Chapter was being mailed.

The next paper presented was on The Engineering Control of Some Solvent Hazards in War Industries, by S. C. Rothman, New York (see p. 319). Owing to the absence of Captain Rothman on military duty, an abstract of the paper was presented by Carl H. Flink, Technical Secretary of the Society.

The meeting adjourned at 11:45 a.m.

President Downs presided at the fourth and final session. He announced the registration of 340 at 2:00 p.m. and anticipated that the attendance would reach 350 before the session had been concluded.

The first paper on Some Effects of Attic Fan Operation on Comfort, by W. A. Hinton and W. G. Wanamaker, was presented in abstract by Professor Hinton (see p. 371).

Owing to the inability of the authors to be present, the next paper on Train Piston Action Ventilation and Atmospheric Conditions in Chicago Subways was prepared by W. E. Rasmus and Edison Brock (see p. 385) and was presented in abstract by Mr. Flink.

As no discussion was offered, President Downs introduced E. R. Ambrose, New York, who presented an abstract of the paper on the Description and Performance of Two Heat Pump Air Conditioning Systems which he prepared with Philip Sporn (see p. 405).

President Downs announced that the presentation of technical papers had been concluded and K. C. Richmond read the following report of the Resolutions Committee on motion of W. A. Russell, Kansas City, Mo., seconded by E. R. Ambrose, it was adopted unanimously.

### Resolutions

*Whereas*, the Semi-Annual Meeting 1944 of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been an outstanding event in the City of Grand Rapids, Mich., June 18 to 20.

*Therefore*, be it resolved, that an expression of thanks and appreciation be adopted by this meeting and be spread upon the minutes of the Society and copies thereof be transmitted to each of the persons and agencies who have contributed toward making this meeting so enjoyable for the members of the Society who attended.

To H. D. Bratt, president, and C. H. Pesterfield, immediate past president, of the Western Michigan Chapter and the Chapter members for the capable manner in which they fulfilled their positions as hosts.

To Chairman Thomas D. Stafford and his Committee on Arrangements, whose careful planning and outstanding efforts have been shown in every detail of this meeting.

To Mrs. O. D. Marshall and her Committee who did such splendid work in entertaining the ladies.

To the Authors and Panel Speakers at technical sessions for their instructive papers and able presentations.

To the newspapers and trade publications whose columns have given advance notices and daily coverage to the Semi-Annual Meeting.

To the Pantlind Hotel employees who contributed to the success of the meeting and comfort of the members under difficult war conditions.

To the Grand Rapids Chamber of Commerce for their cooperation and assistance.

To the Past Presidents and to our Life Member, L. W. Millis, of Kansas City, for their continued interest and attendance at this meeting.

To Dr. M. M. McGorrill for his very inspiring and timely talk on Human Relations.

To the members of the Society who are in the armed forces and others who are giving their services in many ways for the successful prosecution of the war.

To the Banquet Committee in anticipation of the pleasure that we will derive from the Semi-Annual Dinner tonight and the entertaining talks by Mr. Roger Allen and Prof. E. C. Prophet, and finally

To every member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS who has attended this meeting for his loyalty and support, and

Be it further resolved, that the members of this Society pledge their support to the Government in its National Fuel Efficiency Program to expedite the war effort by helping to prevent the needless waste of heat and power.

Respectfully submitted,  
RESOLUTIONS COMMITTEE,  
K. C. Richmond, Chairman.  
R. L. Blanding.  
E. C. Evans.

The meeting was adjourned at 3:30 p.m.

### RESEARCH

Research activities played an important part in the Grand Rapids meeting, for, in addition to the meeting of the Committee on Research on Tuesday afternoon, eleven Technical Advisory Committees also met.

The focal point of research activities was a booklet entitled *Research Projects of Importance to the Industry* which had been prepared on the instructions of the Executive Committee of the Committee on Research. It contained details of 28 research projects suggested by Technical Advisory Committees and approved by the Executive Committee.

#### *Research Committee Meeting*

At a meeting of the Committee on Research which was held on Tuesday afternoon, the Director made a report to the Committee on the activities since the January meeting, presented the financial statement, and requested certain changes in the budget which had become necessary on the removal of the Laboratory to Cleveland and the expanded research program. He reported that all major structural changes to the building at Cleveland had either been made or would be completed within the next three or four weeks, and that a nucleus staff had been engaged and had already settled down into a good working team. The Laboratory had been equipped with the necessary tools, benches, etc., and it was now fully prepared to commence work on actual research investigations.

#### *Research Memorial Library*

The Committee on Research approved the proposal to establish at the Research Laboratory the John R. Allen Memorial Library, the intention being that all of the volumes in the library should bear a suitably prepared bookplate indicating that they were part of this memorial library. The library would be set up to commemorate John R. Allen, the first director of the Research Laboratory, who died while in its service, and who is affectionately remembered by many of the older members of the Society.

### GET-TOGETHER LUNCHEON

A Get-Together Luncheon was held in the Grille Room at 1:00 p.m. with H. D. Bratt, president of Western Michigan Chapter, acting as chairman.

The principal speaker at the luncheon was Dr. M. M. McGorrell, conductor of the Faith for America radio program on a number of Michigan stations, whose subject, *What Makes an Engineer Tick*, proved to be a very interesting outline of the factors which enable engineers to work effectively with their associates and with those who may be in their charge.

## BANQUET

The Semi-Annual dinner started promptly at 7:00 p.m. in the ballroom of The Pantlind, with the singing of the *Star Spangled Banner*, and the invocation was given by Dr. M. M. McGorrell.

T. D. Stafford, Chairman of the Committee on Arrangements, welcomed the 250 members and guests and introduced Brig. Gen. W. A. Danielson, U.S.A., for the presentation of a tribute to the 369 Society members in the armed services.

A tribute to the armed forces was given by Brig. Gen. W. A. Danielson.

Fellow members and guests: It is fitting that we should pause for a moment, as the beginning of the end of the most terrible war in the world's turbulent history, to pay respect to those who have changed from citizens to active members of the armed forces. To me, it is an especial honor that I am privileged to voice this tribute. As a regular Army officer my status is different from the civilian soldier, but perhaps my long service makes me feel more keenly the justification of this tribute. I am humbled by my inability to find words that will adequately express this tribute. My heart is overflowing, and yours, too, I know, and words do not satisfy.

Members in uniform of this Society of ours are doing their share wherever the uniform is worn; many of these brothers, sons, fathers, in the steaming jungles of Burma, in Italy, on the deserts of Iraq, on malaria-infested Guadalcanal, and still others have lived so far through the battle astride the Potomac. Whether it be battleship or battleplane, tank or submarine, anti-aircraft gun or laundry unit, in the fighting lines or the supply echelons, we find members of the Society doing more than their bit, as would be expected.

Each one of the almost 400 members in service—about 15 per cent, one out of every six or seven on our roster—is entitled to be named individually. I owe it to them, but you have already included their names in the Society Journal, which is more permanent than my fleeting mention could be.

In going over this list I came across the names of old friends of yours and mine, and have heard something of them from you. The former president of the Kansas City Chapter, D. D. Zink, is a Lieutenant Colonel on General Staff duty. Captain H. K. McCain is now in the Quartermaster Corps. A. E. Reif is the Fighting Colonel of the 74th Infantry. Captain A. E. Stacey, Jr., controls the flow of materials for destroyers and scouters and landing craft in the Navy. Captain T. H. Werner is in the Signal Corps. Sergeant A. E. Kurtz, who was for seven years with the Laboratory in Pittsburgh, is now on the windswept and isolated Aleutians. J. K. Gonzales is a pilot in the Air Corps. Dr. M. B. Ferderber is now a captain in the Medical Corps. In the Canadian and British Forces, doing their bit, among others, are Lieutenant Colonels J. B. Bishop, J. H. Fox, L. W. Norfolk, and Bombardier Harry Richardson.

The various uniforms have brought the war to this meeting. I know that you all enjoyed talking with our former Director of the Laboratory, Commander Ferry C. Houghten. Commander T. H. Urdahl has left, but Lieutenant-Commander E. A. Queer, Tom's righthand man, is still here. You listened to the interesting paper by Major Brandt. Lieutenant Jack Everetts just returned from the Southwest Pacific.

And now I come to a tribute for service that I wish were otherwise. Captain John Pryke, of the British Army, was made a prisoner of war in North Africa. Two of our brother allies have made the supreme sacrifice: Wing Commander Phil H. Foster, R.C.A.F. and Flying Officer H. L. Temple of the Royal Air Force, were killed in action. We extend our sympathy to those near and dear to them, but we are proud of them. They gave their all that our cause might prevail.

Yes, we sincerely pay our tribute to those in the armed forces, but it must be more than words. We must each do our part in solving the problems here at home that will follow the end of hostilities. Remember, always, that we are first American citizens and then engineers.

It was in large part the failure of the German business man to enter into government, or politics, as we call it, that brought Hitler into power and this war to us, a vacant chair in many homes, perhaps your home. The foundation of this power was laid just after Versailles in the dissatisfaction of the young German Army officers, Major and below, that had returned to no future at home, and often starvation. They had the disintegrating comparison with the years of position, ample food and steady salary in wartime.

Unless we solve the problems ahead wisely and soundly, this sincere tribute to our armed forces may be scorned tomorrow. We each of us must not fail him or our country. With God's help, may we keep it the Land of the Free, and thus pay lasting tribute to our members of the armed forces.

Chairman Stafford introduced the toastmaster Roger Allen, Grand Rapids architect and well-known columnist on the Grand Rapids Press.

#### PRESENTATIONS

J. F. McIntire, Detroit, past president of the Society, was introduced and presented on behalf of the Society the President's Memory Book to Past President Blankin, who acknowledged it with a brief word of appreciation.

Prof. L. G. Miller, East Lansing, Mich., a member of the Council, was introduced, and on behalf of the Western Michigan Chapter, presented the Past President's Plaque to C. H. Pesterfield for his services to and leadership of the Chapter.

Toastmaster Allen explained confidentially that, "there isn't much difference between an architect and an engineer. An architect gets things wrong and an engineer gets things wrong too, but an engineer can do it faster because he uses a slide rule."

#### SOME FINE POINTS ON GEOGRAPHY

Prof. Edward C. Prophet, Associate Professor of Geography at Michigan State College, who has given 500 radio broadcasts and over 350 talks in the past two and one half years on the subject *Geography in the News* was introduced. With the aid of a series of maps the speaker explained that he hoped to give a somewhat different viewpoint on the war and present a means of interpreting the war news that is now being published. He explained that the present war activities are being covered so extensively that one of the greatest difficulties is to interpret the information. He also explained that this gives the greatest possible opportunity for propaganda.

# PROGRAM SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

PANTLIND HOTEL, GRAND RAPIDS, MICH.

June 19-20, 1944

Sunday—June 18

- 10:00 A.M. Guide Publication Committee Meeting, J. F. Collins, Jr., *Chairman*  
(Room 127)
- Finance Committee Meeting, B. M. Woods, *Chairman* (Room 128)
- 1:30 P.M. Council Meeting (*Sadler Lounge*)
- 2:00 P.M. Research Technical Advisory Committee Meetings  
(See Bulletin Board for Committee List)
- 9:30 P.M. Reception and Buffet

Monday—June 19

- 8:30 A.M. REGISTRATION (*Mezzanine*)
- 9:30 A.M. BUSINESS SESSION—(*Ballroom*)—*President* S. H. Downs, presiding  
Greetings—H. D. Bratt, *President* Western Michigan Chapter  
Response by *President* S. H. Downs  
Amendments to By-Laws and Research Regulations  
Selecting Winter Design Temperatures, by Paul D. Close  
Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey  
and L. T. Wright, Jr.  
A Method of Heating a Corrugated-Iron Coal Preparation Plant,  
by E. K. Campbell
- 12:15 P.M. Get-Together Luncheon (*Grille Room*)  
Greetings—Hon. George W. Welsh, *Mayor* of Grand Rapids  
Address—What Makes an Engineer Tick, by Dr. M. M. McGorrrill
- 1:30 P.M. Ladies leave Pantlind Hotel for Grand Rapids Furniture Museum,  
followed by visit to Baker Furniture Exhibition
- 2:00 P.M. TECHNICAL SESSION—(*Ballroom*)—*Vice-Pres.* A. J. Offner, presiding  
Address—Education—The Engineers Major Job in Heating and Fuel  
Conservation—K. C. Richmond  
Panel Discussion—Fuel Conservation—M. F. Blankin, *Chairman*
- 4:00 P.M. Research Technical Advisory Committee Meetings  
(See Bulletin Board for Committee List)  
Committee on Rating Heavy Duty Furnaces—E. K. Campbell, *Chairman*
- 4:00 P.M. Inspection Trips
- 5:30 P.M. Chapter Delegates' Conference (*Amber Suite*)
- 7:00 P.M. Dutch Treat Party (*Grille Room*)

Tuesday—June 20

- 8:30 A.M. REGISTRATION (*Mezzanine*)
- 9:30 A.M. TECHNICAL SESSION—(*Ballroom*)—*Vice-President* C.-E. A. Winslow,  
presiding  
The Engineering Control of Some Solvent Hazards in War Industries,  
by S. C. Rothman  
Industrial Exhaust Ventilation in Industrial Hygiene, by A. D. Brandt  
The Use of Glycol Vapors of Air Sterilization and the Control  
of Air Borne Infection, by B. H. Jennings, Edward Bigg and  
F. C. W. Olson  
Air Disinfection in Ventilation, by W. F. Wells

- 10:30 A.M. Ladies leave Pantlind Hotel for Brunch-Bridge at Cascade Hills Country Club
- 12:00 P.M. Nominating Committee Meeting (*Amber Suite*)
- 2:00 P.M. TECHNICAL SESSION—(*Ballroom*)—President S. H. Downs, presiding  
Some Effects of Attic Fan Operation on Comfort, by W. A. Hinton and W. G. Wanamaker  
Train Piston Action Ventilation and Atmospheric Conditions in Chicago Subways, by W. E. Rasmus and Edison Brock  
Description and Performance of Two Heat Pump Air Conditioning Systems, by Philip Sporn and E. R. Ambrose
- 3:00 P.M. Rug Demonstration
- 4:00 P.M. Committee on Research Meeting—G. L. Tuve, *Chairman* (*Amber Suite*)
- 7:00 P.M. Semi-Annual Dinner (*Ballroom*)  
Toastmaster—Roger Allen  
Tribute to Members in Armed Forces  
Address—Geography in the News—by Prof. Edward C. Prophet, Michigan State College  
Presentation of the Past President's Memory Book to M. F. Blankin

Wednesday—June 21

A.M. Fishing Trip

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**1254**

## SELECTING WINTER DESIGN TEMPERATURES

By PAUL D. CLOSE,\* CHICAGO, ILL.

THERE appears to be no universally accepted rule for selecting winter design temperatures. Obviously the design temperatures should be low enough to provide for all reasonably severe conditions but not necessarily for the lowest temperatures likely to be encountered which may be of short duration, and rarely, if ever, repeated. Such low temperature valleys can generally be bridged over by the heat capacity or fly-wheel effect of the structure. It is of course a simple matter to *play safe* by selecting a design temperature well below that likely to be encountered in the locality of the building. A heating plant designed on this basis would of course be ample, but it would also be more than ample, that is, it would be over-sized. Therefore, the problem is to select a temperature sufficiently low, but yet not too low.

While there is certainly no objection to providing a reasonable reserve or factor of safety in selecting a heating plant, it would seem that some rational or uniform basis for establishing the proper design temperature for each locality would be desirable. It is, of course, recognized that wind velocities must also be considered where the infiltration losses are determined by the crack method, but the present discussion is concerned only with design temperatures.

One rule which has been used to a considerable extent is to base the design temperature on the lowest temperature ever recorded in the locality, the temperature selected being a specified number of degrees—usually 10 to 15—above this lowest recorded temperature. The principal objection to this method is that such minimum temperatures oftentimes are not *normal* in the sense that they are of short duration or might seldom, if ever, be repeated, thus resulting in inconsistent design temperatures.

Whatever method or rule is used for selecting winter design temperatures, it must necessarily be more or less arbitrary, since the rule must be predicated on certain arbitrary assumptions. It would appear that any rule which may be agreed upon must necessarily be approximate because of the nature of the problem. Furthermore, it is also probable that any such rule will involve the usual exceptions and will necessitate adjustments in some cases for local conditions.

In an endeavor to establish some reasonably reliable and consistent formula for selecting winter design temperatures which would eliminate the effect of temporary extremes, a study was made of weather data compiled in various forms by various sources. One particularly helpful and useful compilation of data used in this study was that entitled, *An Analysis of Winter Tem-*

\* Technical Secretary, Insulation Board Institute. Member of A.S.H.V.E.  
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1944.

peratures for One Hundred and Twenty Cities.<sup>1</sup> These data for example, included a tabulation for the various cities of daily *mean* as well as daily *minimum* temperatures. The number of times the various temperatures occurred during a 23-year period (whether mean or minimum) was shown, thus making it possible readily to analyze the temperatures from the standpoint of winter design. Weather Bureau records were also examined and particular attention was given to various *averages* or *normals*.

In many localities, particularly in the larger cities, winter design temperatures have become more or less arbitrarily established through experience, taking cognizance perhaps of Weather Bureau records. For example, it is

TABLE 1—DIFFERENCES BETWEEN JANUARY NORMAL TEMPERATURES AND DESIGN TEMPERATURES GENERALLY USED IN VARIOUS UNITED STATES CITIES

CITY	JANUARY NORMAL†	DESIGN TEMPERATURE IN GENERAL USE‡	DIFFERENCE
Bismarck, N. D.....	7.8	-30	37.8
Huron, S. D.....	12	-25	37
Minneapolis, Minn.....	13.1	-20	33.1
Green Bay, Wis.....	15.6	-20	35.6
Pocatello, Idaho.....	25.7	-10	35.7
Chicago, Ill.....	23.9	-10	33.9
Springfield, Ill.....	26.9	-10	36.9
Cleveland, Ohio.....	26.8	-5	31.8
Pittsburgh, Pa.....	30.9	-5	35.9
Washington, D. C.....	33.9	0	33.9
Oklahoma City, Okla.....	37	0	37
Fort Smith, Ark.....	39.2	5	34.2
Birmingham, Ala.....	46	10	36
Atlanta, Ga.....	43.1	10	33.1
Shreveport, La.....	47.2	10	37.2
Savannah, Ga.....	51.8	15	36.8
Mobile, Ala.....	51.6	20	31.6
New Orleans, La.....	54.7	20	34.7
Jacksonville, Fla.....	55.6	25	30.6

† From U. S. Weather Bureau.

‡ See A.S.H.V.E. GUIDE 1942, pages 130-131.

common practice to use a temperature of -10 F for Chicago and this design temperature has given satisfactory results over a period of years. Similarly -20 F is generally used for Minneapolis and St. Paul, zero degrees for Washington, D. C., +10 F for Atlanta, Ga., and so on. As a part of this study, the design temperatures generally used in the larger cities were plotted on a map of the United States. The temperatures used were based on data obtained in 1940 from the A.S.H.V.E. Chapters throughout the United States and included temperatures for cities in each chapter area. While the temperature variations appeared to be more or less consistent with variations in the temperature zones, there were many obvious inconsistencies and although it might be a comparatively simple matter to adjust these obvious inconsistencies, the purpose of this study was to endeavor to find a rule which

<sup>1</sup> Compiled by Prof. C. M. Humphreys, Carnegie Institute of Technology and assisted by workers furnished by the W.P.A.

would not only eliminate inconsistencies but which would be applicable to any locality, bearing in mind that in some of the smaller or more remote communities or urban areas, design temperatures have not been established by practice or experience. It is reasonable to assume that many of the temperatures now used—perhaps the majority of them—are substantially correct for the respective localities and may be regarded as criteria.

In studying these data it became apparent that the design temperatures now generally used have a more or less fixed relationship to the January normal, except in the case of the obvious inconsistencies. In other words, the design temperature generally used averaged about 35 deg below the January normal, except along the Pacific Coast, where the departure from the January normal averages about 25 deg. This means that the average minimum departure from the January normal for design purposes is 35 deg (except 25 deg on the Pacific Coast). The smaller deviation from the January normal on the Pacific Coast is no doubt due to the equalizing or tempering effect of the Japanese stream.

Table 1 gives the differentials for a few selected cities (other than Pacific Coast) between the January normals and the design temperatures now generally used. It should be noted that the cities listed represent practically the entire range of design temperature zones for the United States, showing that the departure from the January normal is fairly consistent for all zones rather than for just one section. Since design temperatures are generally specified in even 5 deg, a tolerance of plus or minus 5 deg might be allowed in establishing this rule, thus also permitting some adjustment to provide for local conditions.

Table 2 gives the design temperatures for various cities based on a fixed departure from the January normals. In this table the January normals minus 35 deg (25 deg on the Pacific Coast), are given in Column D. The figures in Column D are adjusted up to +5 deg in Column E, the values in this column being the recommended design temperatures for this method. The adjustment was made to the nearest 5 deg in most cases, the exception being instances where the opposite adjustment (but within 5 deg) appeared to be justified. In only one city—Billings, Montana—was an adjustment made of more than 5 deg. Fig. 1 is a map of the United States showing average January temperatures as compiled by the Weather Bureau.

Still another basis for selecting winter design temperatures is the average annual minimum temperature. (See Fig. 2.) This of course is the average of the minimum temperatures which have occurred once each winter for the period 1899-1938, as is indicated on Fig. 2. For example, the average minimum temperature in Chicago is -8 F and in New York -3 F. Minor adjustments would be made in most cases either to the nearest or next lowest even 5 deg temperature. The average minimum has the advantage that it eliminates the effect of abnormal extremes and it would seem to the author that this is a logical basis for the selection of winter design temperatures.

A comparison of three bases of selecting design temperatures for various United States cities is given in Table 3. Columns C and D show the design temperatures for the lowest recorded temperatures, plus 10 and plus 15 deg respectively. Column E gives the commonly used temperatures for each city—which may not necessarily be regarded as reliable criteria in all cases—and Column F is the recommended design temperatures based on the January

TABLE 2—DESIGN TEMPERATURES FOR VARIOUS UNITED STATES CITIES BASED ON JANUARY NORMAL METHOD

A	B	C	D	E
STATE	CITY	JANUARY NORMAL†	JANUARY NORMAL LESS 35 DEG.‡	ADJUSTED DESIGN TEMPERATURE (BASED ON COLUMN D)
Alabama.....	Birmingham.....	46	11	10
	Mobile.....	51.6	16.6	15
Arizona.....	Flagstaff.....	27.6	-7.5	-10
	Phoenix.....	50.9	15.9	20
Arkansas.....	Fort Smith.....	39.2	4.2	5
	Little Rock.....	42	7	10
California.....	Los Angeles.....	55.1	30.1†	30
	San Francisco.....	50	25†	25
Colorado.....	Denver.....	26	-9	-10
	Grand Junction.....	25.2	-9.8	-10
Connecticut.....	New Haven.....	28.4	-6.6	-5
Dist. of Columbia	Washington.....	33.9	-1.1	0
Florida.....	Jacksonville.....	55.6	20.6	20
Georgia.....	Atlanta.....	43.1	8.1	10
	Savannah.....	51.8	16.8	15
Idaho.....	Lewiston.....	32.7	-2.3	0
	Pocatello.....	25.7	-9.3	-10
Illinois.....	Chicago.....	23.9	-11.1	-10
	Springfield.....	26.9	-8.1	-10
Indiana.....	Evansville.....	36	1	0
	Indianapolis.....	28.4	-6.6	-10
Iowa.....	Dubuque.....	19.0	-16	-15
	Sioux City.....	19.0	-16	-15
Kansas.....	Concordia.....	27.0	-8	-10
	Dodge City.....	29.3	-5.7	-10
Kentucky.....	Louisville.....	34.5	-0.5	0
Louisiana.....	New Orleans.....	54.7	19.7	20
	Shreveport.....	47.2	12.2	10
Maine.....	Eastport.....	20.7	-14.3	-15
	Portland.....	22.4	-12.6	-10
Maryland.....	Baltimore.....	34.3	-0.7	0
Massachusetts.....	Boston.....	28.2	-6.8	-5
Michigan.....	Alpena.....	18	-17.0	-15
	Detroit.....	24.5	-10.5	-10
	Marquette.....	16.0	-19.0	-20
Minnesota.....	Duluth.....	9.3	-25.7	-25
	Minneapolis.....	13.1	-21.9	-20
Mississippi.....	Vicksburg.....	48	13	15
Missouri.....	St. Joseph.....	26.0	-9	-10
	St. Louis.....	31.8	-3.2	-5
	Springfield.....	33.2	-1.8	0
Montana.....	Billings.....	22.6	-12.4	-20
	Havre.....	13	-22	-25
Nebraska.....	Lincoln.....	23	-12	-15
	North Platte.....	23.1	-11.9	-15
Nevada.....	Tonopah.....	30.4	-4.6	-5
	Winnemucca.....	27.9	-7.1	-10
New Hampshire.....	Concord.....	21.2	-13.8	-15
New Jersey.....	Atlantic City.....	33.2	-1.8	0

† From U. S. Weather Bureau.

‡ 25 deg. deducted on Pacific Coast as indicated.

TABLE 2—DESIGN TEMPERATURES FOR VARIOUS UNITED STATES CITIES BASED ON JANUARY NORMAL METHOD—*Continued*

A	B	C	D	E
STATE	CITY	JANUARY NORMAL†	JANUARY NORMAL LESS 35 DEG.‡	ADJUSTED DESIGN TEMPERATURE (BASED ON COLUMN D)
New York.....	Albany.....	24.0	-11	-10
	Buffalo.....	24.9	-10.1	-10
	New York City.....	30.9	-4.1	-5
New Mexico.....	Santa Fe.....	29.3	-5.7	-5
North Carolina...	Raleigh.....	41.7	6.7	10
	Wilmington.....	47.2	12.2	15
North Dakota...	Bismarck.....	7.8	-27.2	-30
	Devils Lake.....	2.2	-32.8	-30
Ohio.....	Cleveland.....	26.8	-8.2	-10
	Columbus.....	29.1	-5.9	-5
Oklahoma.....	Oklahoma City.....	37.0	2.0	0
Oregon.....	Baker.....	24.9	-10.1	-10
	Portland.....	39.1	14.1‡	15
Pennsylvania....	Philadelphia.....	32.9	-2.1	0
	Pittsburgh.....	30.9	-4.1	-5
Rhode Island....	Providence.....	27.7	-7.3	-5
South Carolina...	Charlottesville.....	50.2	15.2	15
	Columbia.....	46.5	11.5	10
South Dakota....	Huron.....	12.0	-23.0	-25
	Rapid City.....	22.5	-12.5	-15
Tennessee.....	Knoxville.....	38.8	3.8	0
	Memphis.....	41.3	6.3	5
Texas.....	El Paso.....	44.9	9.9	10
	Fort Worth.....	46.1	11.1	10
	San Antonio.....	52.6	17.6	15
Utah.....	Modena.....	26.8	-8.2	-10
	Salt Lake City.....	29.2	-5.8	-5
Vermont.....	Burlington.....	22.3	-12.7	-15
Virginia.....	Lynchburg.....	37.4	2.4	5
	Norfolk.....	41.4	6.4	5
	Richmond.....	38.2	3.2	5
Washington.....	Seattle.....	39.8	14.8‡	15
	Spokane.....	26.3	-8.7	-10
West Virginia....	Parkersburg.....	33.0	-2.0	-5
Wisconsin.....	Green Bay.....	15.6	-19.4	-20
	La Crosse.....	15.5	-19.5	-20
	Milwaukee.....	20.5	-14.5	-15
Wyoming.....	Lander.....	18.7	-16.3	-20
	Sheridan.....	18.8	-16.2	-20

† From U. S. Weather Bureau.

‡ 25 deg. deducted on Pacific Coast as indicated.

normals as per Column E of Table 2. The average annual minimum temperatures are given in Column G.

While averages may have no particular significance so far as these figures are concerned, it is interesting to note how closely the various averages agree with each other. The average of the values in Column C (lowest temperature +10) is -8.61 F and the average of the values in Column D which of course is 5 deg less (lowest temperature +15) is -3.61 F. The latter

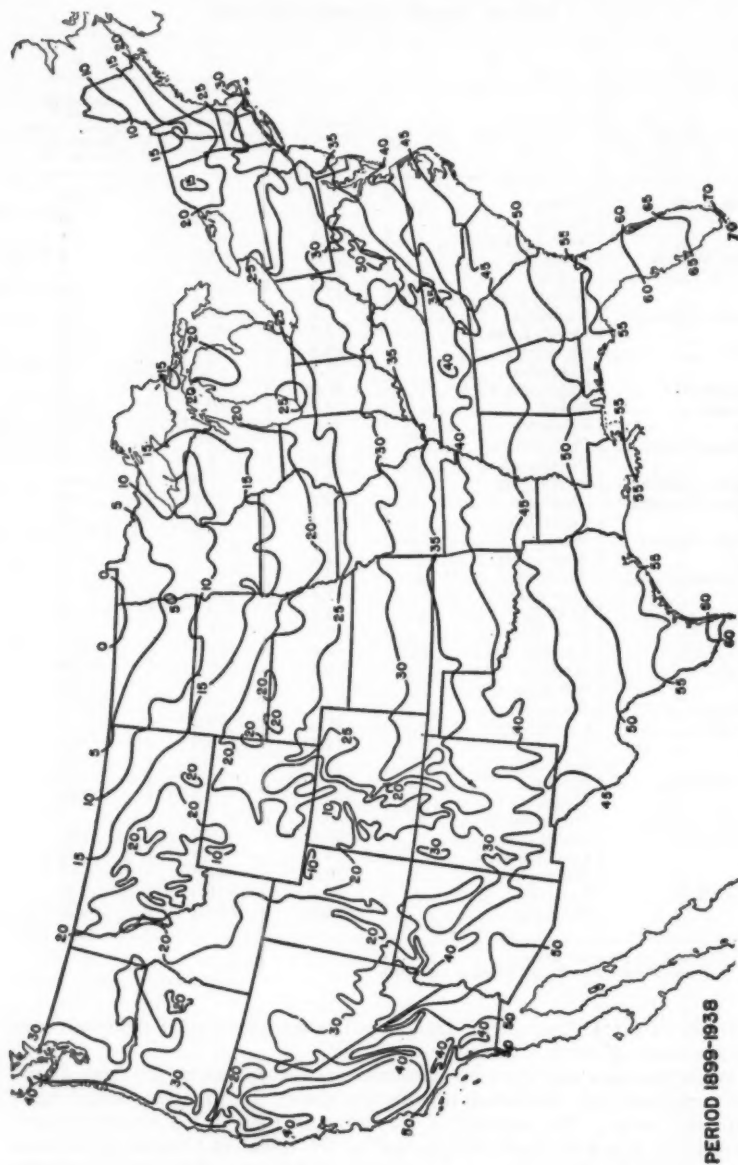


FIG. 1. AVERAGE JANUARY TEMPERATURE



FIG. 2. AVERAGE ANNUAL MINIMUM TEMPERATURE

TABLE 3—COMPARISON OF DESIGN TEMPERATURES BY VARIOUS METHODS

A	B	C	D	E	F	G
STATE	CITY	LOWEST TEMPERA- TURE EVER RECORDED +10 DEG.†	LOWEST TEMPERA- TURE EVER RECORDED +15 DEG.†	DESIGN TEMPERA- TURES COMMONLY USED‡	DESIGN TEMPERA- TURES BASED ON JANUARY NORMAL§	AVERAGE ANNUAL MINIMUM TEMPERA- TURES
Ala....	Birmingham....	0	5	10	10	12
	Mobile.....	9	14	20	15	22
Ariz....	Flagstaff.....	-20	-15	-10	-10	-15
	Phoenix.....	20	25	25	20	26
Ark....	Fort Smith....	-5	0	5	5	6
	Little Rock....	-2	3	10	10	10
Calif....	Los Angeles....	38	43	30	30	37
	San Francisco...	37	42	30	25	37
Colo....	Denver.....	-19	-14	-15	-10	-11
	Grand Junction..	-11	-6	-10	-10	-2
Conn....	New Haven....	-5	0	0	-5	-1
D. C....	Washington....	-5	0	0	0	-1
Fla.....	Jacksonville...	20	25	25	20	29
Ga.....	Atlanta.....	2	7	10	10	12
	Savannah.....	18	23	15	15	22
Idaho...	Lewiston.....	-13	-8	-5	0	1
	Pocatello.....	-18	-13	-10	-10	-12
Ill.....	Chicago.....	-13	-8	-10	-10	-8
	Springfield....	-14	-9	-10	-10	-7
Ind.....	Evansville.....	-6	-1	0	0	1
	Indianapolis....	-15	-10	-10	-10	-6
Iowa....	Dubuque.....	-22	-17	-20	-15	-17
	Sioux City.....	-25	-20	-20	-15	-20
Kans....	Concordia....	-15	-10	-10	-10	-13
	Dodge City.....	-16	-11	-10	-10	-10
Ky.....	Louisville....	-10	-5	-5	0	-5
La.....	New Orleans....	17	22	20	20	26
	Shreveport....	5	10	10	10	16
Maine...	Eastport.....	-13	-8	-10	-15	-15
	Portland.....	-11	-6	-10	-10	-6
Md.....	Baltimore.....	3	8	10	0	8
Mass....	Boston.....	-8	-3	0	-5	-3
Mich....	Alpena.....	-18	-13	-10	-15	-12
	Detroit.....	-14	-9	-10	-10	-11
	Marquette....	-17	-12	-10	-20	-13
Minn....	Duluth.....	-31	-26	-30	-25	-28
	Minneapolis....	-24	-19	-20	-20	-23
Miss....	Vicksburg....	9	14	15	15	18
Mo.....	St. Joseph....	-14	-9	-10	-10	-12
	St. Louis.....	-12	-7	-5	-5	-2
	Springfield....	-19	-14	-10	0	-5
Mont...	Billings.....	-39	-34	-30	-20	-30
	Havre.....	-47	-42	-30	-25	-36
Nebr....	Lincoln.....	-19	-14	-15	-15	-13
	North Platte...	-25	-20	-20	-15	-17
Nev.....	Tonopah.....	-10	0	5	-5	-2
	Winnemucca....	-26	-21	-15	-10	-10
N. H....	Concord.....	-25	-20	-20	-15	-15
N. J....	Atlantic City...	1	6	5	0	6

† Lowest temperature ever recorded up to 1943.

‡ See A.S.H.V.E. GUIDE, 1942, pages 130-131.

§ From Column E, Table 2. (January normal minus 35 F, except 25 F on Pacific Coast, and adjusted.)

TABLE 3—COMPARISON OF DESIGN TEMPERATURES BY VARIOUS METHODS—Continued

A	B	C	D	E	F	G
STATE	CITY	LOWEST TEMPERA- TURE EVER RECORDED +10 DEG.†	LOWEST TEMPERA- TURE EVER RECORDED +15 DEG.†	DESIGN TEMPERA- TURES COMMONLY USED‡	DESIGN TEMPERA- TURES BASED ON JANUARY NORMAL§	AVERAGE ANNUAL MINIMUM TEMPERA- TURES
N. Y....	Albany.....	-14	-9	-5	-10	-11
	Buffalo.....	-10	-5	0	-10	-4
	New York.....	-4	1	0	-5	-3
N. M....	Santa Fe.....	-3	2	0	-5	0
N. C....	Raleigh.....	8	13	15	10	13
	Wilmington.....	15	20	20	15	18
N. D....	Bismarck.....	-35	-30	-30	-30	-31
	Devils Lake.....	-36	-31	-30	-30	-33
Ohio....	Cleveland.....	-7	-2	-5	-10	-2
	Columbus.....	-10	-5	-10	-5	-3
Okla....	Oklahoma City.....	-7	-2	0	0	2
Ore....	Baker.....	-15	-10	-15	-10	-17
	Portland.....	8	13	10	15	18
Pa.....	Philadelphia.....	-1	4	0	0	6
	Pittsburgh.....	-10	-5	-5	-5	-2
R. I....	Providence.....	-7	-2	0	-5	1
S. C....	Charleston.....	17	22	15	15	22
	Columbia.....	8	13	10	10	19
S. D....	Huron.....	-33	-28	-25	-25	-26
	Rapid City.....	-24	-19	-20	-15	-21
Tenn....	Knoxville.....	-6	-1	0	0	2
	Memphis.....	1	6	0	5	9
Texas....	El Paso.....	5	10	0	10	16
	Fort Worth.....	2	7	0	10	12
	San Antonio.....	14	19	10	15	21
Utah....	Modena.....	-22	-17	-15	-10	-15
	Salt Lake City.....	-10	-5	-10	-5	2
Vt.....	Burlington.....	-19	-14	-20	-15	-17
Va.....	Lynchburg.....	3	8	10	5	8
	Norfolk.....	12	17	15	5	15
	Richmond.....	7	12	10	5	10
Wash....	Seattle.....	13	18	15	15	20
	Spokane.....	-20	-15	-15	-10	-5
W. Va....	Parkersburg.....	-17	-12	-10	-5	-1
Wis....	Green Bay.....	-26	-21	-20	-20	-18
	La Crosse.....	-33	-28	-25	-20	-21
	Milwaukee.....	-15	-10	-10	-15	-12
Wyo....	Lander.....	-30	-25	-25	-20	-26
	Sheridan.....	-31	-26	-25	-20	-27

† Lowest temperature ever recorded up to 1943.

‡ See A.S.H.V.E. GUIDE, 1942, pages 130-131.

§ From Column E, Table 2. (January normal minus 35 F, except 25 F on Pacific Coast, and adjusted.)

compares favorably with the averages for Columns E and F, which are -3.81 F and -3.75 F, respectively. The average of the Column G figures is approximately -2 F.

The A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions has recently decided that the design dry-bulb temperature for winter should be taken as that temperature which is equalled or exceeded for 97½ per cent of the hours during the months of December, January, February

and March. It will be interesting to see how design temperatures selected on this basis compare with temperatures selected on other bases.

Reiterating the statement previously made, whatever method is decided upon must necessarily be arbitrary, and it should be added, such a method to be generally acceptable should not produce results which will deviate greatly from the general practice in the majority of localities. For this reason, it is the author's opinion that the average annual minimum temperature (Column G, Table 3), provides a reasonably acceptable basis.

## DISCUSSION

H. D. JAMES,<sup>2</sup> Pittsburgh, Pa. (WRITTEN): This paper brings up the practical application of design temperature. Some engineers may take the difference between 70 deg inside and -5 deg outside in the Pittsburgh area and use this figure to calculate the heat loss. This will give the maximum size of central heating unit required with its corresponding size of pipes or ducts, etc. The size of this equipment can be reduced if the auxiliary heating equipment is taken into consideration. In most of the cities, such as Pittsburgh, having available natural gas, the residences have gas grates or other types of gas heaters in the rooms to give local heat when the central plant is not operating or when the room is cold early in the morning or at other times. Those of us who burn coal or coke do not start the central plant until the cold weather becomes settled. In my own house, it is not necessary to start the furnace until the outside temperature drops below 40 deg. The furnace can be operated to give a general house temperature of 60 or 65 deg in severe weather and the room temperature increased by operating the local gas heater.

When this method is intelligently applied, the total gas and coke bill is less than when obtaining all the heat from the coke furnace. The gas grate adds a local comfort from its radiant heat and is therefore frequently used even when it is not necessary to raise the room temperature.

Pittsburgh temperature seldom reaches zero and then only for a few days. Even a low temperature of 10 deg does not last long. If a design temperature difference of 65 deg is used instead of 75 deg, the size of the central heating plant can probably be reduced without discomfort to the persons using the residence.

Much depends upon the house design. Zero temperature is usually accompanied by high north or northwest winds much in excess of the design velocity used to calculate the infiltration heat losses. Unless the windows are unusually tight and the side walls well insulated, the rooms on the windward side of the house cannot be made comfortable even with an oversized heating plant until the wind goes down.

A common sense approach should be taken to the problem of heating during the few extreme days of cold weather to avoid installing too large a central heating plant. During extreme cold weather, the gas pressure usually goes down and a central gas heater will not deliver its rated output. Infiltration heat losses are the big error in all heating calculations and, as the house gets older, infiltration increases. Zero weather with a strong northwest gale blowing is usually uncomfortable inside as well as outside, but we find ways to endure it for the few days that it lasts.

C. E. BENTLEY, San Francisco, Calif.: I observed in the information compiled that both California stations, Los Angeles and San Francisco, in Column E, are 30 deg, whereas Column F has Los Angeles 30 deg and San Francisco 25 deg. I feel that probably 30 deg is more equitable for San Francisco. Where 35 deg is generally used as winter design temperature. I wish to call attention to many valley towns which are more extreme in heat and cold, which are not covered at the present time by this tabulation.

<sup>2</sup> Consulting Engineer, Pittsburgh, Pa.

T. H. URDAHL, Washington, D. C.: The author's basis for selection of winter design temperatures approaches the more logical basis for selection of winter design temperature than any that we have had presented so far. As Chairman of the Research Technical Advisory Committee on Winter Design Conditions, we began a study of the existing weather design data a good many years ago, and in orienting ourselves as to what sort of job we would have to do, we started with bare fundamentals of analysis of temperatures.

We found that there was one important factor that had heretofore been omitted, the factor of time. For how long a period of time does a given low temperature pertain? As in the design of an automobile, are you designing a machine to run continually at 90 mph, or should it be able to attain that speed occasionally?

Therefore, what should the high point of efficiency of the equipment be, for its maximum speed, or for that speed over which it operates most of the time? That analysis applies to a heating plant or any other system.

In our analysis we found that averages were important, and that is the direction along which the committee is at present proceeding with its analysis. We do find one thing, however: there is no simple, easy, fast means of re-analyzing and accurately setting up our design condition data in as complete a form as it should be available to the engineer today. It is a very difficult, tedious, boring and expensive job, to analyze the millions of readings which must be so analyzed in order to get a fineness that I think these times deserve in engineering calculations.

Just a word about that term *fineness*. Our basic data, upon which we have originally based temperature selections and design, was compiled for the purposes of furnishing the farmer with crop weather data, and, accordingly, the accuracy of the information, the instrumentation applied to it, was not nearly of the degree of fineness which is really necessary when the figure is used in engineering calculations.

Today, in obtaining weather readings for flying, a high degree of accuracy is necessary and has been attained. It is the temperature data, the meteorological data that is taken for our air lines and for people who are flying, that we are today using as a basis of our analyses, rather than the old data which we found, on going through weather bureau data, was a very much hit-or-miss proposition.

I want to congratulate the author, as it is the first paper on the subject that has been brought before the Society since the data presented by Mr. Albright.<sup>3</sup>

We have not kept the subject as alive as it should be. It is something that is fundamental, that is very important in all our work, and everyone should have an interest in the job that your Committee on Research is now doing.

It is rather a large undertaking, the setting up of new design data for both summer and winter use.

W. E. K. MIDDLETON,<sup>4</sup> Toronto, Ont., Canada: There is a very important paragraph in this paper which should be emphasized. The author states that he has not taken wind conditions into account, which is understandable because of the complications that this would involve. In Northwestern Canada particularly, and it occurs, to a certain extent, in the Rocky Mountain States, there are a great many days during the winter when it is very cold but is also extremely calm. The air is stable up to a height of several thousand feet. As a result, we find in the station buildings that we do not have to heat them nearly to the extent that we should expect, if we had to design for the mean minimum temperatures that you would obtain.

The mean minimum temperature at Fort Smith in the Northwest Territories, for example, is somewhat below -50 F, and yet I am sure that a building at Fort Smith

<sup>3</sup> Analysis of Summer Weather Data in the United States, by J. C. Albright. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 397.)

<sup>4</sup> Department of Transport, Toronto, Ontario, Canada.

is no harder to heat than a building in Montana. I doubt if it is as hard; because nearly all the time, when it is  $-50^{\circ}\text{F}$  at Fort Smith, it is absolutely dead calm, and therefore the heat transfer is less. That does not happen in the East to nearly the same extent. When we have a cold spell in Toronto it is always blowing very hard, and, consequently, I do not think that the design temperature for Toronto is any too low.

I am not a heating engineer, but I have a feeling that it is not low, because I do not know many people in Toronto whose houses are really warm when it is really cold.

I feel that the investigations of the TAC Committee, on Weather Design Conditions, are very important, and should take the wind into consideration, even if that is a pretty difficulty thing to do.

**AUTHOR'S CLOSURE:** Mr. Bentley referred to Los Angeles and San Francisco and the difference of 30 and 25—30 for Los Angeles and 25 for San Francisco. That was based on the January normal basis, which was more or less discarded in favor of one given in Column G, the average annual minimum.

Referring to Commander Urdahl's remarks, the Technical Advisory Committee on Weather Design Conditions is, of course, working on this problem, and it is hoped that they will come out with the right answer; but it seems to me, nevertheless, that there are two things about this situation. First, no matter what method is adopted, it must be somewhat arbitrary, because it must be predicated on certain assumptions.

The other thing that occurs to me is that whatever method is agreed upon must correspond in general with present accepted practice; otherwise, I am afraid it will not be too well accepted.

Mr. Middleton referred to the question of wind conditions. Of course, as we all know, the heating engineer does take wind conditions into consideration. Theoretically, so far as design conditions are concerned, the ideal situation would be to take a combination of temperature and wind velocity, namely, that combination of temperature and wind velocity which produced maximum calculated heat loss; but, that would be different for every building. That has been considered in the past. In fact, about 10 years ago I presented a paper on that subject dealing with concurrent combinations of temperature and wind velocity.

However, it appears to be too complicated for application to most problems. One combination of temperature and wind velocity would affect one building in one way and another building in another way, depending upon the amount of window crack and so on. But wind velocity nevertheless must be and is taken into consideration, as we all know, in calculating heat losses.



**1255**

## PERIODIC HEAT FLOW—HOMOGENEOUS WALLS OR ROOFS

By C. O. MACKEY\* AND L. T. WRIGHT, JR.,\*\* ITHACA, N. Y.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Cornell University.

THIS REPORT summarizes the work done during 1943 on unsteady heat flow through homogeneous building materials. The general method followed is that explained in a previously published paper,<sup>1</sup> but a brief summary of the method is included in this report. Ranges chosen for the variables are those suggested by the A.S.H.V.E. Research Technical Advisory Committee on Cooling Load in Summer Air Conditioning. More complete results of the study have been presented to this Committee in the form of preliminary reports.

### VARIABLES, SYMBOLS, AND UNITS

The variables involved in this study of periodic heat flow are listed in Table 1 with their symbols, units, and suggested ranges.

### SOL-AIR TEMPERATURE<sup>3</sup>

" In the first paper by Mackey and Wright,<sup>3</sup> a new term was introduced which is important in the study of weather data and solar heat gain—*equivalent temperature* of outdoor air. Since the term *equivalent temperature* has been widely used for many years in England to give a basis for estimating comfort conditions the authors accepted the suggestion of an alternative term *sol-air temperature* and have used it throughout this paper. Sol-air temperature will be understood to mean the temperature of the outdoor air, which in contact with a shaded building surface, would give the same rate of heat transfer and the same temperature distribution through the material as exists with the actual dry-bulb temperature of the outdoor air and the actual intensity of solar radiation incident upon that surface. For either steady or un-

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<sup>1</sup> Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 148.)

<sup>2</sup> This discussion applies only to those building materials which do not directly transmit solar radiation.

<sup>3</sup> Loc. Cit. See Note 1.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Grand Rapids, Mich., June 1944.

steady flow of heat, the sol-air temperature for a value of the outdoor air film coefficient of heat transfer of 4 Btu/hr ft<sup>2</sup> F is:

$$t_o = t_a + \frac{bI}{4} \quad (1)$$

where

$t_a$  = the dry-bulb temperature of the outdoor air, Fahrenheit.

$I$  = the intensity of solar radiation incident upon the outdoor surface, Btu/hr ft.<sup>2</sup>

$b$  = solar absorptivity of surface.

For example, if the intensity of solar radiation incident upon a building surface with a solar absorptivity of 0.4 is 200 Btu/hr ft<sup>2</sup> the effect of the solar radiation upon the rate of heat transfer through the material and upon the

TABLE 1—VARIABLES, SYMBOLS, UNITS AND SUGGESTED RANGES

QUANTITY	UNITS	SYMBOL	RANGE SUGGESTED BY COMMITTEE
Inside air film coefficient of heat transfer.	Btu/hr ft <sup>2</sup> F	$h_o$	1.65 (fixed) ✓
Outdoor air film coefficient of heat transfer	Btu/hr ft <sup>2</sup> F	$h_L$	4.00 (fixed) ✓
Thermal conductivity.....	Btu/hr ft F	$k$	0.00833 to 28 ✓
Volumetric specific heat.....	Btu/ft <sup>3</sup> F	$\rho c$	1 to 60
Thickness of material.....	ft	$L$	0.0104 to 3
Absorptivity of outside surface for solar radiation.....	none	$b$	0; 0.4; 0.7 (selected values)

distribution of temperature in that material is precisely as if the temperature of the outdoor air were 20 F higher with the surface receiving no solar radiation. In this example, the sol-air temperature would be 20 F higher than the shaded dry-bulb temperature of the air.<sup>11</sup>

The equation which fits the periodic sol-air temperature as a function of time is of the general form:

$$t_o = t_m + \sum_{n=1}^{\infty} t_n \cos(15n\theta - a_n) \quad (2)$$

where

$t_m$  = the daily average sol-air temperature, F.

$\theta$  = the time, measured in hours after noon (at noon,  $\theta = 0$ ; at 1 P.M.,  $\theta = 1$ , etc.)

$n$  = the harmonic coefficient ( $n = 1$  for first harmonic, or fundamental;  $n = 2$  for second harmonic, etc.)

$t_n$  = the harmonic temperature coefficient, Fahrenheit ( $t_1$  for first harmonic, etc.).

$a_n$  = the harmonic phase angle, degrees, ( $a_1$  for first harmonic, etc.).

#### TEMPERATURE OF INSIDE SURFACE OF BUILDING

The exact solution is complete for finding the temperature of the inside surface of a *homogeneous building material* at any time for the case where (1) the incident solar radiation and temperature of outdoor air are periodic, (2) the temperature of the indoor air is constant, (3) the outdoor air film coefficient of heat transfer is 4.0, and (4) the indoor air film coefficient of heat transfer is 1.65 Btu/hr ft<sup>2</sup> F.

For this case, at any time  $\theta$ , the temperature of the inside surface of the building material is:

$$t_0 = t_1 + \frac{0.606(t_m - t_1)}{0.856 + \frac{L}{k}} + \sum_{n=1}^{\infty} \lambda_n t_n \cos(15n\theta - a_n - \phi_n) \quad (3)$$

where

$t_0$  = the temperature of the inside surface of the building material, Fahrenheit.

$t_1$  = the constant temperature of the indoor air, Fahrenheit.

$L$  = the thickness of the material, ft

$k$  = the thermal conductivity of the material, Btu/hr ft Fahrenheit.

$\lambda_n$  = the harmonic decrement factor ( $\lambda_1$  for first harmonic, etc.).

$\phi_n$  = the harmonic lag angle, degrees ( $\phi_1$  for first harmonic, etc.).

The instantaneous rate of heat transfer from the indoor surface, in Btu/hr is:

$$q = 1.65 A(t_0 - t_1) \quad (4)$$

where

$A$  = the area of the inside surface at temperature  $t_0$ , ft<sup>2</sup>.

Equations for the decrement factor,  $\lambda$ , and for the lag angle,  $\phi$ , were given in the original paper and are repeated in Appendix A. Graphs are presented from which these values may be read. Values of the decrement factor are read from Fig. 1, while Fig. 2 gives values of the lag angle. For the  $n$ th harmonic of the *actual* material, use the fundamental (or first harmonic) of a fictitious or *equivalent* material; the equivalent material designated by the subscript  $e$ , must then have the same thermal conductivity and thickness as the actual material but a volumetric specific heat which is  $n$  times that of the actual material; or

$$\left(\frac{k}{L}\right)_e = \frac{k}{L} \quad (5)$$

while

$$(k\rho c)_e = nk\rho c \quad (6)$$

A numerical illustration of the exact solution is given in Appendix B.

It is recognized that few engineers will care to use a method as complex as the accurate solution. An approximate method is next explained which is simple enough to warrant general use without too great sacrifice in the accuracy of the final result.

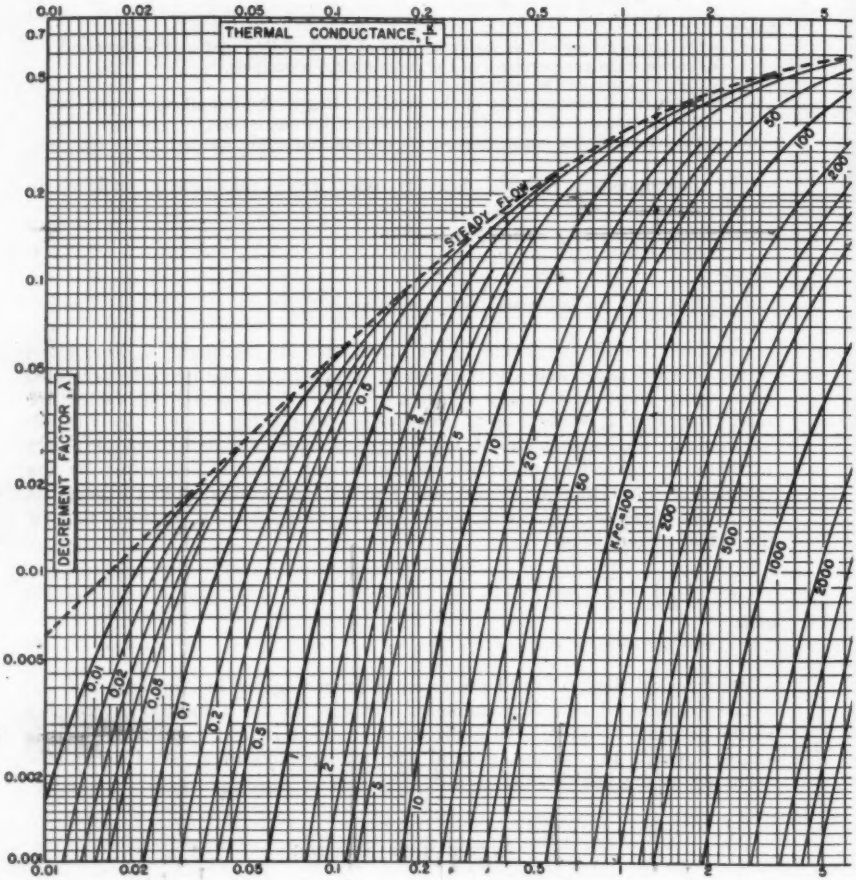
#### APPROXIMATE SOLUTION

An approximate solution is presented for the temperature of the inside surface of the material which gives acceptable results and which starts with a knowledge of the periodic sol-air temperature as a function of time.

Let the steady-flow mean daily temperature of the inside surface of the material be  $t_M$ , where

$$t_M = t_1 + \frac{0.606(t_m - t_1)}{0.856 + \frac{L}{k}} \quad (7)$$

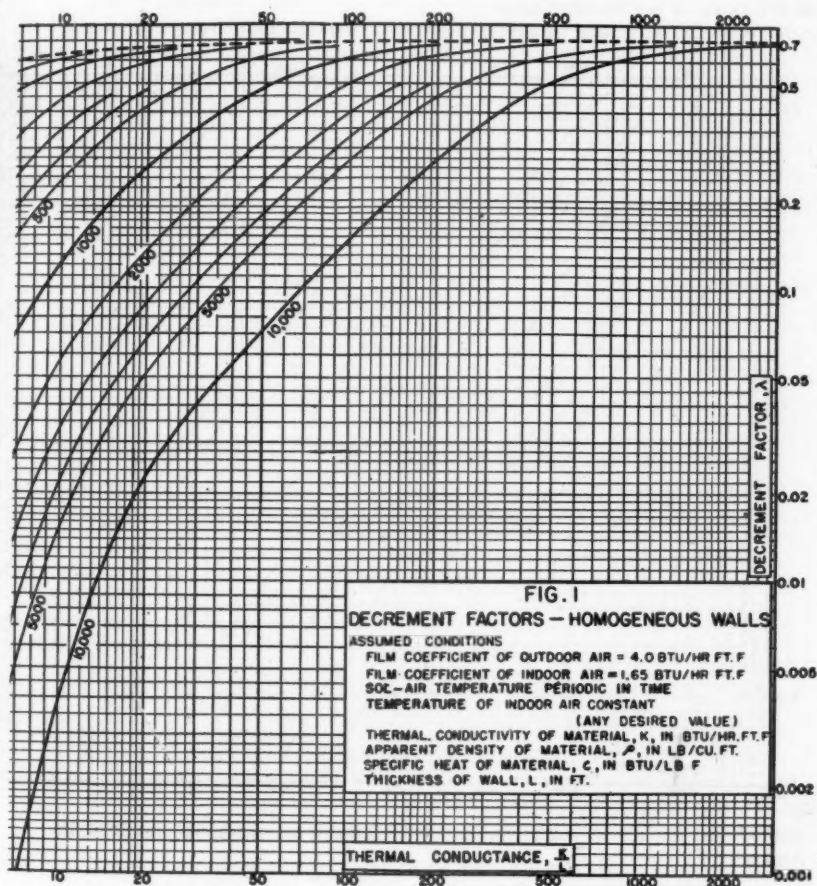
With two assumptions explained in Appendix C, it may be shown that Equation 8 gives the temperature of the inside surface of the material in terms of the sol-air temperature at a time which is earlier by the fundamental time lag, in hours,  $(\phi_1/15)$ .



The temperature of the inside surface of the material at a time  $(\theta + \frac{\phi_1}{15})$  hours after noon is related to the sol-air temperature at a time  $\theta$  hours after noon as follows:

$$(t_s)_{\theta + \frac{\phi_1}{15}} = t_m + \lambda_1 [(t_s)_\theta - t_m] \dots \dots \dots (8)$$

This equation states that (approximately) the sol-air temperature at any time affects the temperature of the inside surface of the material at a time which is  $\frac{\phi_1}{15}$  (the fundamental time lag) hours later. It implies that the shape

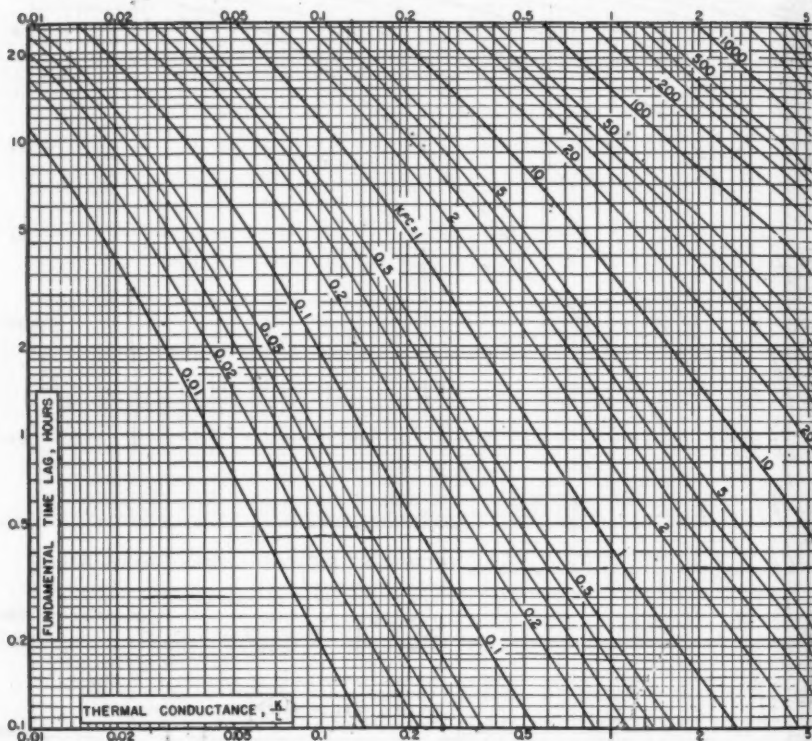


of the periodic curve of surface temperature *vs.* time is the same as the shape of the period sol-air temperature *vs.* time curve, but reduced in range and displaced in phase.

In preliminary reports, the results obtained for the variation with time of the temperature of the inside surface of widely different building materials in different thicknesses by both the exact and approximate methods have been compared. The shape of the temperature *vs.* time curve and the time of

maximum heat transfer from the inside surface have been found to be reproduced fairly well by the approximate method. The maximum rate of heat transfer found by the approximate method has always been found to be on the safe side—greater than that given by the exact method. The greatest percentage error in the heat transfer rate has always been found to occur at the times of the smallest rates.

A second approximate method which gives more accurate results than the



method just explained, but at the cost of a little more work, is presented in Appendix C.

#### SUMMARY OF RESULTS

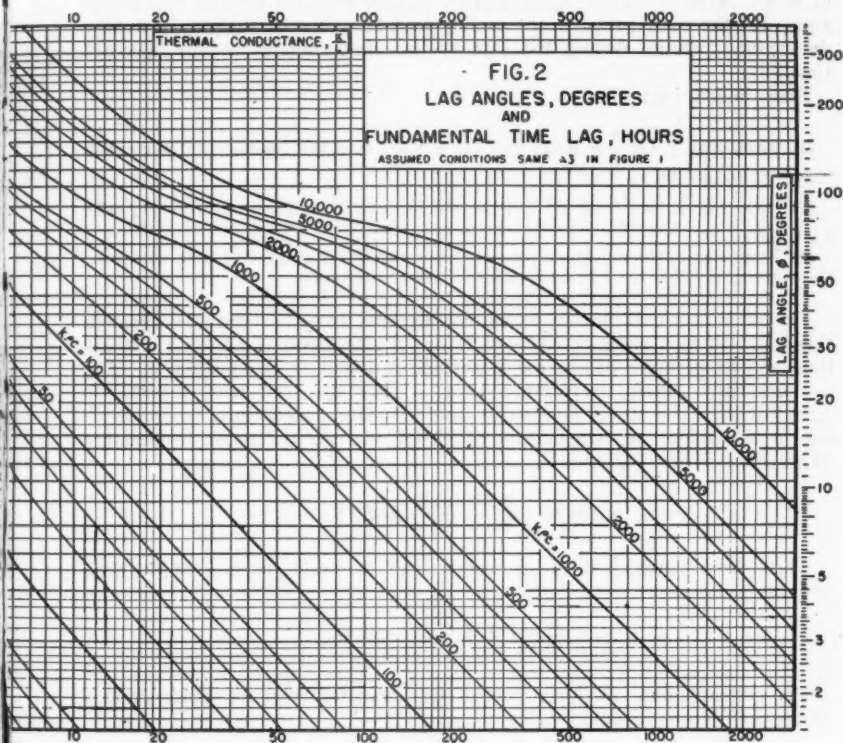
The assumptions that have been made in this study follow:

- (1) The temperature of the outdoor air and the total solar radiation incident upon the building surface have been assumed to be cyclic with a period of 24 hours.
- (2) The temperature of the indoor air has been assumed to be held constant during the period of 24 hours.
- (3) The building wall or roof has been assumed to be made of a single, homogeneous material.
- (4) The rate of heat transfer from the outdoor air to the building surface has been

assumed constant and equal to 4 Btu/hr ft<sup>2</sup> F (outside air film coefficient of heat transfer; summer conditions).

(5) The rate of heat transfer from the inside surface of the structure has been assumed constant and equal to 1.65 Btu/hr ft<sup>2</sup> for each degree of temperature difference between surface and indoor air.

The information which must be known in order to find the contribution



to the cooling load due to heat transfer from the inside surface of the building material at any time of day includes:

- (1) The periodic sol-air temperature *vs.* time curves for the particular locality, orientation and solar absorptivity of the building surface (design curves).
- (2) The thickness, thermal conductivity, and volumetric specific heat of the wall or roof and the solar absorptivity and orientation of the building surface.
- (3) The constant temperature of the indoor air.

#### EXAMPLE OF RECOMMENDED PROCEDURE (APPROXIMATE METHOD OF SOLUTION)

- (1) Assume the sol-air temperature *vs.* time curve for the locality under consideration to be given by Fig. 3. (These design curves should be based upon weather data.)

(2) Assume the unshaded wall under consideration faces west and is made of homogeneous red brick with a thickness of 8 in., a thermal conductivity of 5 Btu in./hr ft<sup>2</sup> F, a volumetric specific heat of 20 Btu/ft<sup>3</sup> F, and a solar absorptivity of the exterior surface of  $b = 0.7$ .

(3) Assume the constant temperature of the indoor air to be 80 F.

It is required to find the maximum contribution to the cooling load due to heat transfer from the inside surface of this wall, the time of this maximum, and the contribution to the cooling load at a time of 3 p.m. (Actually, the method permits finding the rate of this heat transfer at any specified time.)

Steps in solution:

(1) From Fig. 1, for  $\frac{k}{L} = 0.625$  and  $k\rho c = 8.33$ , read the fundamental decrement factor,  $\lambda_1$ , to be 0.15.

(2) From Fig. 2, for  $\frac{k}{L} = 0.625$  and  $k\rho c = 8.33$ , read the fundamental time lag to be 5.5 hour.

(3) From Fig. 3, the maximum sol-air temperature for a wall facing west with solar absorptivity of 0.7 is 133.2 F, and occurs 3.7 hours after noon.

The maximum contribution to the cooling load from heat transfer through this wall will then occur at  $(3.7 + 5.5)$  or 9.2 after noon (9.12 p.m.) At this time, the temperature of the inside surface is found from Equation 8. The average daily sol-air temperature (Fig. 3) for the west wall,  $b = 0.7$ , is  $t_m = 91.6$  F. The steady-flow mean inside surface temperature (corresponding to an outdoor air temperature of 91.6 F and an indoor air temperature of 80 F) is, from Equation 7,

$$t_{si} = 80 + \frac{0.606(11.6)}{0.856 + 1.6} = 82.9 \text{ F}$$

Then, from Equation 8, the temperature of the inside surface of the wall at 9:12 p.m. is:

$$t_o = 82.9 + 0.15(133.2 - 91.6) = 89.1 \text{ F}$$

*at 9:12 70 has been added to surface*

The corresponding maximum rate of heat transfer from the inside surface of the wall, from Equation 4, is:

$$\frac{q}{A} = 1.65(89.1 - 80) = 15.0 \text{ Btu/hr ft}^2$$

The next part of the example is to find the rate of heat transfer at 3 p.m. The sol-air temperature which influences this rate of heat flow is that temperature at a time 5.5 hours (fundamental time lag) earlier than 3 p.m. or at 2.5 hours before noon (9:30 a.m.). From Fig. 3, the sol-air temperature at this time is found to be 87.8 F. Then, from Equation 8, at 3 p.m., the temperature of the inside surface of the wall is:

$$t_o = 82.9 + 0.15(87.8 - 91.6) = 82.3 \text{ F}$$

*steady flow 9:30 a.m.*

The corresponding contribution to the cooling load at 3 p.m. is:

$$\frac{q}{A} = 1.65(82.3 - 80) = 3.8$$

Btu/hr ft<sup>2</sup>

It will be noted that this recommended solution uses the approximate method and that the method will give the rate of heat transfer at any specified time of day.

#### EFFECT OF VALUES OF AIR FILM COEFFICIENTS

Although Figs. 1 and 2 are *exactly* correct only when the outdoor air film coefficient of heat transfer is 4.0 and the indoor air film coefficient of heat transfer is 1.65 Btu/hr ft<sup>2</sup> F, values of the decrement factor and lag angle read from these graphs are approximately correct for the usual departures from these values. This is discussed further in Appendix D.

#### APPENDIX A

Equations for Decrement Factor  $\lambda$  and Lag Angle  $\phi$ .  
Decrement factor:

$$\lambda = \sqrt{\frac{2}{F^2 + G^2}} \quad \dots \dots \dots (A1)$$

Lag Angle

$$\phi = \tan^{-1} \left( \frac{F - G}{F + G} \right) \quad \dots \dots \dots (A2)$$

where

$$F = (\pi_1 + 1) C_1 + \frac{C_3}{\pi_3} + 2\pi_1 \pi_3 C_4$$

$$G = (\pi_1 + 1) C_2 + \frac{C_4}{\pi_3} - 2\pi_1 \pi_3 C_3$$

$$C_1 = \cos \pi_2 \cosh \pi_2 + \sin \pi_2 \sinh \pi_2$$

$$C_2 = \cos \pi_2 \cosh \pi_2 - \sin \pi_2 \sinh \pi_2$$

$$C_3 = \sin \pi_2 \cosh \pi_2$$

$$C_4 = \cos \pi_2 \sinh \pi_2$$

$$\pi_1 = \frac{h_o}{h_L} \text{ (fixed at 0.4125 for Figs. 1 and 2).}$$

$$\pi_2 = sL$$

$$\pi_3 = \frac{ks}{h_o}$$

$$s = \sqrt{\frac{0.1309 \rho c}{k}}$$

The decrement factors and lag angles given in Figs. 1 and 2 are the fundamental or first harmonic values. For the harmonics of order  $n$ , use

$$\pi_{2n} = \sqrt{n} \pi_{21}$$

and

$$\pi_{3n} = \sqrt{n} \pi_{31}$$

## APPENDIX B

A numerical illustration of the exact method of solution follows:

Assumed sol-air temperature series for west wall ( $b = 0.7$ ):

$$\begin{aligned}
 t_a = & 91.6 + 23 \cos(15\theta - 51) \\
 & + 9.1 \cos(30\theta - 92) \\
 & + 5.9 \cos(45\theta - 166) \\
 & + 3.6 \cos(60\theta - 217) \\
 & + 1.0 \cos(75\theta - 292) \\
 & + 0.93 \cos(90\theta - 68) \dots\dots\dots (A3)
 \end{aligned}$$

Assume an unshaded west wall made of homogeneous red brick with a thickness of 8 in., a thermal conductivity of 5 Btu in./hr ft<sup>2</sup> F, a volumetric specific heat of 20 Btu/ft. F, and a solar absorptivity of the exterior surface of 0.7. For this wall,

$$\frac{h}{L} = 0.625 \text{ Btu/hr ft}^2\text{F},$$

and

$$k\rho c = 8.33 \text{ Btu}^2/\text{hr ft}^3\text{F}^2$$

Values of the decrement factor  $\lambda$  and of the lag angle  $\phi$  are shown in Table 2.

TABLE 2—VALUES OF DECREMENT FACTOR AND LAG ANGLE

$h/L = 0.625$				
HARMONIC	$n$	$(k\rho c)_n$	$\lambda_n$	$\phi_n$
First (fundamental).....	1	8.33	0.150	83
Second.....	2	16.7	0.080	131
Third.....	3	25.0	0.048	166
Fourth.....	4	33.3	0.030	195
Fifth.....	5	41.7	0.020	220
Sixth.....	6	50.0	0.014	242

For an assumed constant temperature of the indoor air of  $t_i = 80$  F, the temperature of the inside surface of the wall at any time  $\theta$  is found from Equation 3, Equation A3, and these tabular values as:

$$\begin{aligned}
 t_o = & 80 + 2.86 + 3.45 \cos(15\theta - 134) + 0.729 \cos(30\theta - 223) \\
 & + 0.283 \cos(45\theta - 332) + 0.108 \cos(60\theta - 412) \\
 & + 0.020 \cos(75\theta - 512) + 0.013 \cos(90\theta - 310)
 \end{aligned}$$

At a time of 8 P.M. ( $\theta = 8$ ), for example, the temperature of the inside surface of this wall is:

$$\begin{aligned}
 t_o = & 80 + 2.86 + 3.35 + 0.70 + 0.25 + 0.04 + 0.00 + 0.01 \\
 = & 87.21 \text{ F}
 \end{aligned}$$

At this time, the instantaneous rate of heat transfer from the inside, or room surface, of this wall is found from Equation 4 to be:

$$\frac{q}{A} = 1.65 (87.21 - 80) = 11.90 \text{ Btu/hr ft}^2.$$

The approximate method of solution yields a result for the maximum rate of heat transfer from the inside surface of this wall which is about 25 per cent higher than the actual maximum rate, a maximum temperature of the inside surface of the wall about 1.9 F higher than the actual maximum temperature, and a time of this maximum about

one hour later than the time of the actual maximum. The same curve for sol-air temperature (Fig. 3) is assumed for both cases.

### APPENDIX C

The actual temperature of the inside surface of the material at a time  $\theta$  hours after noon is:

$$(t_o)_\theta = t_M + \sum_{n=1}^{\infty} \lambda_n t_n \cos(15n\theta - a_n - \phi_n) \dots\dots\dots (C1)$$

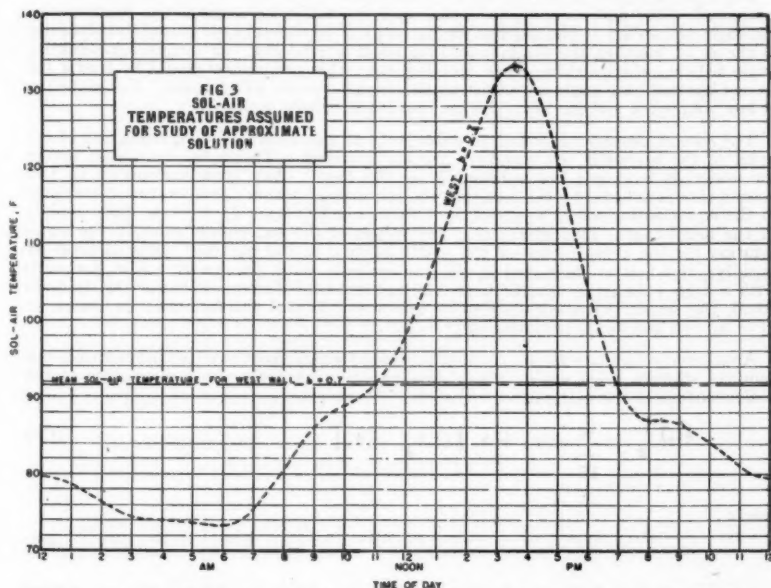


FIG. 3. SOL-AIR TEMPERATURES ASSUMED FOR STUDY OF APPROXIMATE SOLUTION

The fundamental time lag, in hours, is the fundamental lag angle divided by 15; the temperature of the inside surface of the material, at a time  $\theta + \frac{\phi_1}{15}$  hours after noon is:

$$\begin{aligned} (t_o)_{\theta + \frac{\phi_1}{15}} &= t_M + \sum_{n=1}^{\infty} \lambda_n t_n \cos[15n(\theta + \frac{\phi_1}{15}) - a_n - \phi_n] \\ &= t_M + \sum_{n=1}^{\infty} \lambda_n t_n \cos[15n\theta - a_n - (\phi_n - n\phi_1)] \dots\dots\dots (C2) \end{aligned}$$

The first assumption made is that:

$$\phi_n = n\phi_1 \dots\dots\dots (C3)$$

The second assumption made is that:

$$(1 - \lambda_n) = (1 - \lambda_1) \dots\dots\dots (C4)$$

With these two assumptions, it may be shown that the temperature of the inside surface of the material at a time  $\theta + \frac{\theta_1}{15}$  hours after noon is related to the sol-air temperature at a time  $\theta$  hours after noon as follows:

$$(t_s)_\theta + \frac{\theta_1}{15} = t_m + \lambda_1 [(t_s)_\theta - t_m] \quad \dots \quad (C5)$$

The reasonableness of the two assumptions made in the approximate solution may be examined. The first assumption is that each harmonic lag angle is the fundamental lag angle multiplied by the order of the harmonic. In the example given in explanation of the exact solution, the fundamental lag angle was 83 deg, the second harmonic lag angle 131 deg, the third harmonic 166 deg, etc.; this assumption would be true if the second harmonic lag angle were 166 deg, the third harmonic 249 deg, etc. For materials of low volumetric specific heat, this assumption is closely met.

The second assumption states, in effect, that all decrement factors, regardless of the order of the harmonic, are equal for a given material. This is far from true, except for materials having a low volumetric specific heat. Any error introduced by this assumption is reduced by the fact that the successive harmonic temperature coefficients in the equation for sol-air temperature (the  $t_n$ 's in Equation 2) decrease quite rapidly in the usual case. This means that the values of the successive terms in the series solution for the temperature of the inside surface of the building material drop off rapidly; this fact is illustrated in the one example given previously. Also, for walls with a high volumetric specific heat, the fundamental decrement factor ( $\lambda_1$ ) is quite small, the second harmonic decrement factor ( $\lambda_2$ ) is still smaller, etc.; in this case  $(1 - \lambda_1)$  is a good approximation of  $(1 - \lambda_n)$ .

Another approximate method is next presented which gives more accurate values but requires slightly more work. The following equation gives the temperature of the inside surface of the material:

$$(t_s)_\theta + \frac{\theta_1}{15} = t_m + \lambda_0 [(t_s)_\theta - t_m] \quad \dots \quad (C6)$$

where

$$\lambda_0 = \frac{\lambda_1 + \lambda_2}{2} \quad \dots \quad (C7)$$

It will be noted that the only difference between Equation C6 and Equation C5 is the decrement factor,  $\lambda$ , used. In Equation C5, the fundamental (or first harmonic) decrement factor is used, while in Equation C6, the arithmetic mean of the fundamental and second harmonic decrement factor is used.

When this latter method is followed, the fundamental decrement factor is read from Fig. 1 when the values of  $k/L$  and  $k_{pc}$  are known for the actual material and thickness; the second harmonic decrement factor is then read from the same figure for the same value of  $k/L$ , but the equivalent value of  $k_{pc}$  which is twice the value of  $k_{pc}$  for the actual material (see Equations 5 and 6). For the 8-in. brick wall, these decrement factors are  $\lambda_1 = 0.15$  and  $\lambda_2 = 0.08$ . By using  $\lambda_0 = 0.115$ , a more accurate estimate of the rate of heat transfer from the inside surface may be obtained. For the same data as used in the recommended procedure, the approximate maximum temperature of the inside surface of the brick wall facing west is, (from Equation C6),

$$t_o = 82.9 + 0.115 (133.2 - 91.6) \\ = 87.7 \text{ F}$$

The corresponding maximum rate of heat transfer from the inside surface of this wall is:

$$\frac{q}{A} = 1.65 (87.7 - 80) = 12.7 \text{ Btu/hr ft}^2$$

This result is closer to the exact series solution of Appendix B which gave 11.9 Btu/hr ft<sup>2</sup> for this rate than the simpler approximate method which gave a result of 15.0 Btu/hr ft<sup>2</sup>.

#### APPENDIX D

The effect of using values of the outdoor and indoor film coefficients of heat transfer other than the assumed values of 4 and 1.65 Btu/hr ft<sup>2</sup> F, respectively, may be demonstrated by examples. Calculated values of the fundamental decrement factor, the

TABLE 3—CALCULATED VALUES OF FUNDAMENTAL DECREMENT FACTOR, LAG ANGLE, AND TIME LAG FOR 8-INCH BRICK WALL

OUTDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER	INDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER	FUNDAMENTAL DECREMENT FACTOR	FUNDAMENTAL LAG ANGLE, DEGREES	FUNDAMENTAL TIME LAG, HR
$h_L$	$h_o$	$\lambda_1$	$\phi_1$	$\phi_1/15$
4.0	1.65	0.150	83	5.5
4.0	0.50	0.234	97	6.5
4.0	1.50	0.158	84	5.6
3.0	1.50	0.146	87	5.8
2.0	1.50	0.126	92	6.1

fundamental lag angle, and the fundamental time lag are shown in Table 3 for the 8-in. brick wall of the previous examples ( $k = 0.417$  Btu/hr ft F;  $\rho c = 20$  Btu/ft<sup>3</sup> F).

With the outdoor air film coefficient of heat transfer held constant, a decrease in the value of the indoor air film coefficient slightly increases the time lag and appreciably increases the fundamental decrement factor. The effect is to increase the daily range in temperature of the inside surface; the maximum temperature of that surface is raised, and the minimum temperature is lowered. With the indoor air film coefficient of heat

TABLE 4—EFFECT OF FILM COEFFICIENT CHANGES UPON MAXIMUM RATE OF HEAT TRANSFER FROM INSIDE SURFACE

OUTDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER	INDOOR AIR FILM COEFFICIENT OF HEAT TRANSFER	MAXIMUM RATE OF HEAT TRANSFER FROM INSIDE SURFACE, (APPROXIMATE METHOD) BTU/HR FT <sup>2</sup>	TIME OF MAXIMUM RATE OF HEAT TRANSFER (APPROXIMATE METHOD)
$h_L$	$h_o$	$q/A$	$p.m.$
4.0	1.65	15.0	9:12
4.0	0.50	7.9	10:12
4.0	1.50	14.5	9:18
3.0	1.50	16.9	9:30
2.0	1.50	20.6	9:48

transfer held constant, a decrease in the value of the outdoor air film coefficient slightly increases the time lag but decreases the fundamental decrement factor. The effect is to decrease the daily range in temperature of the inside surface; the maximum temperature of that surface is lowered and the minimum temperature raised. At the same time, any departure of the outdoor air film coefficient from the assumed value of four changes the sol-air temperature at any time when sun is shining on the building surface; a decrease in this coefficient will raise the sol-air temperature.

The complete effect of these changes in film coefficients upon the maximum rate of heat transfer from the inside surface of the 8-inch brick wall facing west is shown in Table 4.

The indoor air film coefficient of heat transfer is a combined coefficient of convective and radiant heat transfer. For large vertical surfaces, the film coefficient of heat transfer by natural convection is  $0.27 (\Delta t)^{0.25}$ ; for a temperature difference of 10 F, this coefficient is 0.48 Btu/hr ft<sup>2</sup> F. For warm horizontal surfaces facing down, this film coefficient is  $0.2 (\Delta t)^{0.25}$  or 0.16 Btu/hr ft<sup>2</sup> F for a temperature difference of 10 F. The radiant heat exchange at the interior surface is complicated; for an interior surface with an emissivity of unity which exchanges heat with other black surfaces at air temperature, the rate of radiant heat loss from the warm surface is, closely, 1.1 Btu/hr ft<sup>2</sup> for each degree of temperature difference from surface to air. If all the other interior surfaces of an enclosure are at the same temperature as the inside surface of an exterior wall or roof, or if the inside surface of this exterior wall or roof is covered with a material having an emissivity of zero, there can be no radiant heat loss from that surface. With common non-reflecting inside surfaces, the combined film coefficient of heat transfer for vertical walls in still air is probably between 1.5 and 1.6 Btu/hr ft<sup>2</sup> F, provided that the temperature of all other surfaces in the enclosure is the same as the temperature of the air; for horizontal roofs under similar conditions, this combined film coefficient is slightly lower. In this report, Figs. 1 and 2 are based upon values of the outdoor and indoor air film coefficients of heat transfer of 4.0 and 1.65 Btu/hr ft<sup>2</sup> F, respectively. As far as the indoor film coefficient of heat transfer is concerned, it is believed that these decrement factors and lag angles are sufficiently exact for non-reflecting inside surfaces seeing other interior surfaces at air temperature. In support of this contention, note that with the 8-in. brick wall of the example, a change in the value of  $h_o$  from 1.65 to 1.50 lowers the maximum rate of heat transfer at the inside surface from 15.0 to 14.5 Btu/hr ft<sup>2</sup>. On the other hand, these figures would not give correct results for exterior walls or roofs lined with a highly reflecting substance or for a case where the other interior surfaces of the enclosure were at the same temperature as the surface in question. Note that a highly reflective lining applied to the inside surface of the brick wall would lower the maximum rate of heat transfer from 15.0 to 7.9 Btu/hr ft<sup>2</sup>.

The value of the outdoor air film coefficient of heat transfer has a multiple effect in the problem. It affects the sol-air temperature and the values of the decrement factor and lag angle. Equation 1 for sol-air temperature is admittedly imperfect in some respects. During the day, the wind velocity changes and the value of the outdoor air film coefficient of heat transfer also changes. Instead of a constant value of  $h_o$  of 4 Btu/hr ft<sup>2</sup> F, a value that depended upon the instantaneous wind velocity should be used for perfection. During clear nights, there is a considerable loss of heat from the exterior wall by radiation to the cold sky; this effect, which would tend to lower the daily mean sol-air temperature, is not considered, but its neglect is on the safe side. Values of the outdoor air film coefficient of heat transfer are commonly based upon the cooperative research<sup>4</sup> between the University of Minnesota and the A.S.H.V.E. Although these tests did not duplicate outdoor air conditions, exactly, they showed that the film coefficient increased with air movement, mean temperature and surface roughness. For a brick surface at a mean temperature of 80 F, for example, values of the film coefficient were found to be about 2 for a parallel air velocity of 0 mph, 4 for an air velocity of 5 mph, and 6 for an air velocity of 10 mph. The last table shows that a decrease in  $h_o$  from 4 to 2 Btu/hr ft<sup>2</sup> F, through the combined effects on sol-air temperature and decrement factor will raise the maximum rate of heat transfer from the inside surface of the 8-in. brick wall from 14.5 to 20.6 Btu/hr ft<sup>2</sup> for  $h_o = 1.5$ , when the result is obtained by the approximate method of solution; also, the time lag is increased by about 0.5 hour. It should be noted, however, that the value of 15 Btu/hr ft<sup>2</sup> for the maximum rate of heat transfer as found from the approximate method, is roughly, 25 per cent higher than the value of 11.9 found by the exact but complex series method in Appendix B for the same wall. This exact series method applied with sol-air temperature and decrement factor corrected to  $h_o = 2$  would give about the same result as the approximate method with  $h_o = 4$ .

The authors believe that the simplest procedure is to use a constant value of  $h_o$  of 4 in the equation for sol-air temperature, regardless of the wind velocity. Then, by using the approximate method to find the maximum rate of heat transfer based

<sup>4</sup> A.S.H.V.E. RESEARCH REPORT No. 869—Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren, and J. L. Blackshaw. (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429.)

upon Figs. 1 and 2 for  $h_i = 4$  and  $h_o = 1.65$ , the results will be very close for the lowest possible outdoor air movement; for a wind velocity of 10 mph, the estimated rate of heat transfer is always on the safe side. Further, the approximate method always gives a slightly greater time lag than the exact method, and the effect of decreased outdoor air movement is in the same direction.

## DISCUSSION

F. E. GIESECKE, College Station, Tex. (WRITTEN): The authors have earned the gratitude of the engineering profession by their splendid work, reflected by this paper

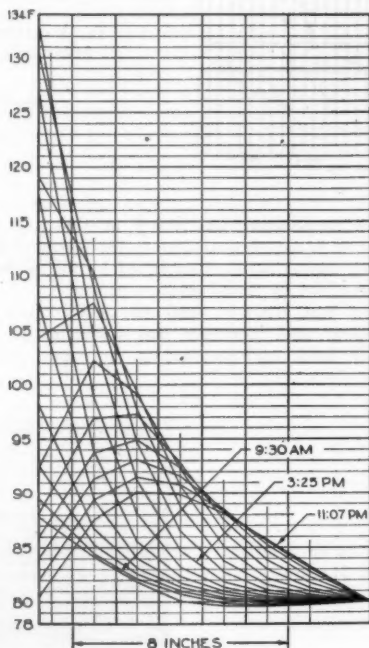


FIG. A. GRAPHICAL ANALYSIS OF HEAT FLOW THROUGH BRICK WALL

and by the paper presented at the January 1943 meeting.<sup>5</sup> Their work will no doubt be continued so as to make the results of their studies of greater value to the practicing engineer.

It is hoped that this brief discussion and the experimental data submitted herewith may be of some assistance in future studies.

If the flow of heat through the 8-in. brick wall, described in the paper, is studied by means of the approximate graphical method, described in a previously published article,<sup>6</sup> the results will be as shown in Fig. A. To employ graphical analysis, it is

<sup>5</sup> Summer Comfort Factors as Influenced by Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 148.)

<sup>6</sup> The Flow of Heat Through Walls, by F. E. Giesecke. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 441.)

necessary to know or to assume the thermal gradient in the wall at the beginning of the study. In this case, it was assumed that the interior surface of the wall had cooled slightly below 80 F during the night and that by 9:30 a.m. the indoor air temperature had risen to 80 F, so that, at the beginning of the study, there was still a slight flow of heat into the 8-in. wall from the indoor air. It was also assumed

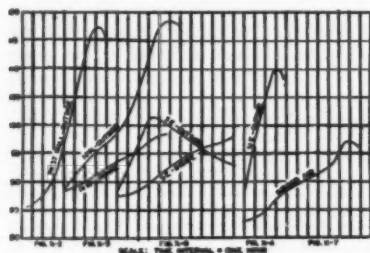


FIG. B. VARIATION IN SURFACE TEMPERATURES OF 8-IN. WALL

that the outdoor sol-air temperature was 87.8 F at 9:30 a.m. and that the form of the thermal gradient at 9:30 a.m. was similar to the forms of the thermal gradients, shown in Fig. 13 of the former article<sup>7</sup> published by the Society, in which thermal gradients had been determined by actual measurements.

To study the flow of heat, the 8-in. wall was divided into 5 slabs, each 1.6 in. thick. The corresponding time increment was calculated to be 51.3 min for the physical qualities of the brick wall specified in the paper. Having determined the

FIG. C. BUILDING WHERE HEAT FLOW STUDIES WERE MADE



time increment and knowing the varying sol-air temperature (Fig. 3) the successive sol-air temperatures were found and the 17 successive thermal gradients drawn in the usual manner. The first thermal gradient shown in Fig. A is for 9:30 a.m. and the last one for 11:07 p.m.

It appears from the diagram that the interior wall surface temperature had risen from about 79.6 F to about 80.5 F or about 0.9 F during the time interval from 9:30 a.m. to 3:25 p.m. This would correspond to a rate of heat flow into the room of about 1.48 Btu per hour if the first thermal gradient had been assumed on the basis of an interior wall surface temperature of 80 F at 9:30 a.m. instead of 79.6 F.

<sup>7</sup> Loc. Cit. Note 6 (see Fig. 13, p. 455).

It also is apparent from the diagram that the maximum rate of heat flow into the room occurred at about 11:30 p.m. and that it is at the rate of about 1.65 (84.5-80.0) or 7.44 Btu per hour; however the increase in heat flow from 9:12 p.m. to 11:30 p.m. is very slight, according to the graphical analysis. These values are somewhat lower than those calculated by the authors; the differences are, no doubt, the results of differences in the methods of analysis.

It also appears from this diagram that the exterior wall surface temperatures of the author's 8-in. wall are as shown in Fig. B(X-2) for the sol-air temperatures shown in Fig. 3 and for the rates of heat flow shown in Fig. A.

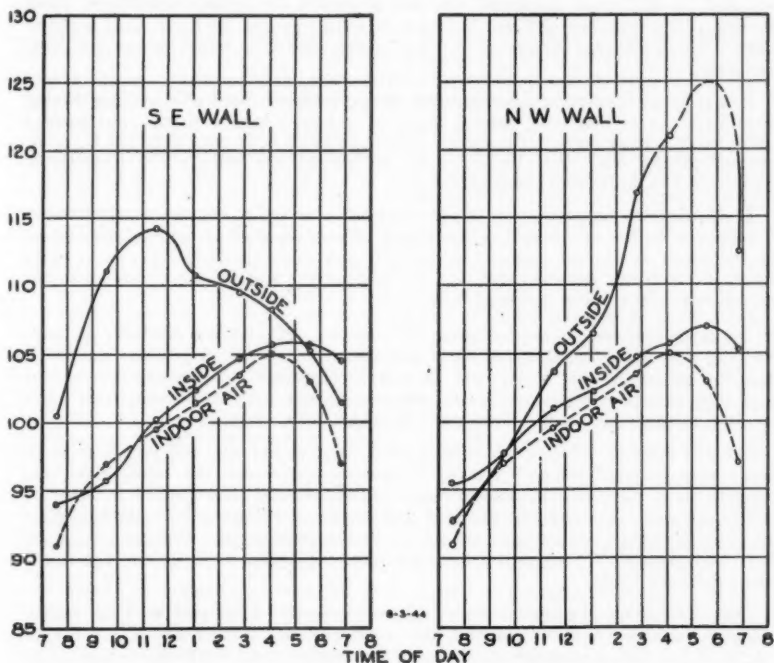


FIG. D. SURFACE TEMPERATURE CHARTS FOR S. E. AND N. W. WALLS

To secure a few experimental data, readings were taken on June 14, 1944, on three of the 8-in. red brick walls of the second story of the building of which the east corner is shown in Fig. C. This building fronts S 38 deg—30 min E. As there were clouds in the sky during part of the time the temperatures recorded are somewhat lower than would be obtained under complete sunshine conditions.

The varying exterior wall surface temperature of the northeast wall of this building is shown in Fig. B(X-4); the varying exterior and interior wall surface temperatures of the northwest wall are shown in Fig. B(X-5). The varying exterior and interior wall surface temperatures of the southeast wall are shown in Fig. B(X-6). The varying indoor air temperature of the space between the northwest and southeast walls is shown in Fig. B(X-7). This space receives its heat primarily through the

roof. The temperature of the southwest wall was not taken because the wall is in the shade of another building.

An opportunity to obtain temperatures on a clear warm day was afforded on August 4, 1944. The results are shown in Fig. D for the S.E. and N.W. walls. Since the thermometer used to determine the outside wall temperature was calibrated only to 120 F it was necessary to estimate the outside wall surface temperature by observation of the vibrations of the needle above 120 F. The estimated values are shown by means of a dotted line.

The surface temperatures shown in Figs. B and D were taken by means of a dermalor supplied by McKesson Appliance Co. and graduated for studies relating to panel heating. The diagrams of Figs. A, B and D were prepared by R. G. Cox, a senior student in Mechanical Engineering. The hours shown in these figures are local sun time.

It may be of interest to note here that during a recent study of a building having exterior walls of limestone, painted black on a base course 4 ft high and painted white on the upper portion, the surface temperatures at 4:00 p.m. were 120 F on the surface painted black, and 106 F on the surface painted white. The temperature of the air 1 in. in front of the wall was 94 F.

C. M. ASHLEY, Syracuse, N. Y.: I would like to say a few words about what I believe to be the significance of this paper. The method which was presented here is the result of several years of trying different other methods. We have tried hydraulic methods and electrical methods of analysis, and we have also attempted to measure heat transfer in actual walls.

All those methods, to some extent, have defects. The present method may seem to be a very highly mathematical and abstract approach to the problem and yet it has the advantage that it is more general in character, and permits the use of any type of wall construction to far better advantage, and, we believe, with much greater accuracy than any of the methods which have preceded it.

We can say that we have a tool here which, with proper use, will permit us to do some very practical things. Where, 15 years ago, we took the steady flow heat transfer as a basis of our summer load calculations and then applied a factor of safety in order to arrive at the values of load, we can now analyze and take advantage of it by knowing in detail what the nature of our problems is. We can insulate to better advantage. We can adjust the capacity of the system to fit the average rather than the peak load.

One other thing, the values of the volumetric specific heat, and, to some extent, also, of the absorptivity of material, are at present very inexact and it is the hope of the A.S.H.V.E. Committee on Research that experimental work can be carried on to obtain more exact values for these. With these data available, we shall have a body of material from which accurate wall transmission data can be obtained.

G. L. TUVE, Cleveland, Ohio: I am glad Mr. Ashley brought up the point that there are a great many methods for analyzing periodic heat flow. Certainly the authors are to be complimented on their success with the method they have used. Perhaps some of you saw the item in the June issue of *Electronics*, reporting that students at M.I.T. had analyzed this problem with electronic devices, and compressed the 24-hour heat-flow cycle into one sixtieth of a second, reading the results with an oscillograph. Our Society has also previously sponsored cooperative research on electrical methods and other methods of heat-flow analysis.

I want to point out just one thing—that whatever method of analysis is used, there are a number of selections or assumptions to be made before the analysis is started. One must select the volumetric specific heat of the material, the conductivity, the

inside surface coefficient, and the outside surface coefficient, to say nothing of the weather curve in terms of the sol-air temperature.

The Society's Committee on Research is aware of the complication involved in this whole problem. There are five projects dealing with periodic heat flow in this list of twenty-eight projects to which Director Tasker has called your attention. I think the Society can do something toward showing what the correct selections or assumptions are, so that the final results of any analysis of heat flow in walls will be more accurate than is otherwise possible.

I want to congratulate the authors on this excellent way of presenting fairly general results in a limited number of charts.

It appears to me that Mr. Ashley in his discussion fails to distinguish between methods of determination and methods of presentation. He refers to attempts of analysis by the hydraulic and the electric methods and by measuring actual walls, but then states that "The present method . . . is more general in character. . . ."

As far as methods of determination are concerned the only legitimate basis for preference between calculation on one hand or use of the hydraulic or electric method on the other are those of accuracy and of economy. The electric method has, of course, an inherent amount of inaccuracy. However, it is believed that this inaccuracy is at least offset by the limitation of the mathematical method used by the authors, who limit themselves to the first harmonic in their analysis.

As regards the presentation, the method selected by the authors has the big advantage of being limited in space. However it involves, as Mr. Ashley concedes, considerable amount of calculation which is easy to carry out but tedious for the practicing engineer not in daily contact with numerical operations. An alternative solution of presentation would be a number of charts from which the desired values could be read directly. Such a method of presentation had been considered at the time when the writer made a preliminary survey<sup>8</sup> of this problem for the Society. This method calls for more space and for more preliminary work but frees the practical man using the results from calculations to which he is not accustomed.

The investigation by measuring heat transfer in actual walls is less accurate than any of the other methods. In order to get satisfactory results it would be necessary to repeat such tests quite frequently and to take average values. These average values then could be used in a presentation of general applicability either following the authors' method or the method of charts previously mentioned.

Professor Tuve mentions the work done at M.I.T. in electric analysis. When building the Heat and Mass Flow Analyzer at Columbia University<sup>9</sup> the possibility of using very much smaller *time-constants* than ultimately selected and measuring by means of an oscillograph had been considered. One disadvantage of this method is to be seen in the impossibility of varying conditions during the experiment, at least with any simple means. Therefore it is believed that working with high resistances and high capacitances in comparatively long values of time is preferable.

**AUTHORS' CLOSURE:** The authors wish to thank those who discussed the paper. Regarding the remarks made by Professor Tuve, it is true that there are a lot of

<sup>8</sup> Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschakis. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 75.)

<sup>9</sup> Determining Unsteady-State Heat Transfer in Solids, by V. Paschakis and H. D. Baker. (*Heat Treating and Forging*, August, 1941.)

Heat Flow Problems Solved by Electrical Circuits, by V. Paschakis. (*Heating, Piping & Air Conditioning*, December, 1941.)

A Method for Determining Unsteady-State Heat Transfer by Means of an Electrical Analogy, by V. Paschakis and H. D. Baker. (A.S.M.E., *Transactions*, February, 1942, No. 2, pp. 105-112.)

Application of an Electrical Model to the Study of Two-Dimensional Heat Flow, by M. Avrami and V. Paschakis. (*Transactions, American Institute of Chemical Engineers*, June 25, 1942, pp. 631-652.)

Loc. Cit. Note 8.

variables in this problem. We have only standardized two—the outdoor film coefficient of heat transfer and the inside film coefficient of heat transfer. We have tried to effect a combination of some by the introduction of sol-air temperature, so that the effects of outdoor temperature and incident solar radiation can be properly combined.

In regard to volumetric specific heat and thermal conductivity, which are the only two thermal properties of the material influencing the unsteady heat flow, it is also true that a short cut may be possible.

The apparent density of the material governs, to a considerable degree, its thermal conductivity ( $k$ ), as you all know, and it also governs to a considerable degree its volumetric specific heat ( $\rho c$ ). The specific heat of most of the common materials on a weight basis—that is expressed in Btu per (pound) (degree Fahrenheit) does not vary widely, regardless of whether the material may be mineral, vegetable, or animal in origin. In other words, the specific heat of a cellulose product is about 0.32 Btu per (pound) (degree Fahrenheit); for rock and rock fibre 0.2; for glass and glass fibre, 0.18. Therefore, on a pound basis, it does not matter much what the origin of the material may be. The product of thermal conductivity and volumetric specific heat depends primarily upon the apparent density in pounds per cubic foot, and a curve of the product of  $k$  and  $\rho c$  plotted *vs.* apparent density would come very close to giving that product for materials both known at the present time and unknown or unused at the present time.



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**1256**

## A METHOD OF HEATING A CORRUGATED IRON COAL PREPARATION PLANT

By E. K. CAMPBELL,\* KANSAS CITY, MO.

**T**HE PROBLEMS involved in heating a coal tipple and coal washing plant built of corrugated iron are so unusual that perhaps a description of a more or less experimental plant may be of interest. It is experimental because there is no way of measuring or guessing intelligently some factors of the problem, but it is a plant intended and guaranteed to meet the



FIG. 1. COAL TIPPLE FOR STRIP MINE

problems to the extent that there should be no freezing of pumps and pipes, a prolific source of trouble in corrugated iron coal preparation plants.

Fig. 1 shows a view of the tipple with the main elevator from the dump pits on the left, and the extension of the building over the tracks on the right, where an uninsulated steel floor is perforated for the car loading chutes, which must be kept open for inspection.

The coal preparation plant belongs to a coal company of Kansas City, which operates strip coal mines in a number of different locations. The experience of the company has shown that a considerable area of coal bearing land can be stripped and the coal exhausted in a comparatively short period. This semi-temporary character of the strip mine makes it necessary at intervals to move the entire plant in order to avoid too long hauls from the shovel to the tipple. This was the immediate occasion of the building of this particular tipple. It was formerly located 14 miles to the south, but that field became exhausted. It was moved to its present location and enlarged in capacity in opening a new field.

\* President, E. K. Campbell Heating Co. Member of A.S.H.V.E.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Grand Rapids, June, 1944.

Because of this necessity for occasional moving, the owners have standardized on corrugated iron buildings with steel frames, thereby making it possible to so design and construct the buildings that they can be taken down, a section at a time, and very quickly moved and re-erected in another location. This construction, of course, creates many problems for the engineer who undertakes to provide heating equipment. Any attempt to heat the building as such, as distinguished from spot heating, must be more or less experimental and dependent on making the building tighter than usual for most of these galvanized buildings.

Hence, any attempt to estimate the heat load involves a very large element of guessing. In this particular case, the area of the corrugated iron and steel surface was estimated and doubled, and proper factors were applied for a

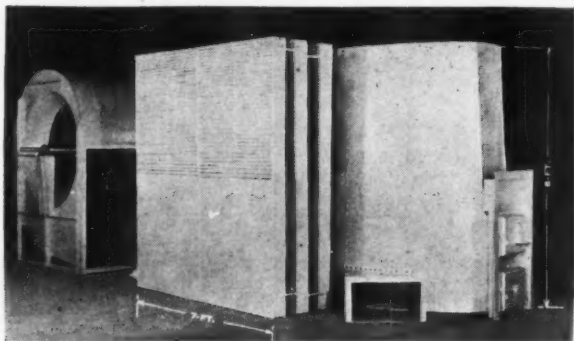


FIG. 2. FAN FURNACE BEFORE INSTALLATION

60 deg temperature rise, to obtain an inside temperature of 40 F at  $-20$  F outside, and to arrive at a heat load of about 3,600,000 Btu.

It is the practice of the author to arrive at a value for air leakage by using a percentage of the heat transmission loss. This has been found to be a highly satisfactory method. In ordinary churches, schools, theaters, and buildings without too many openings, the leakage is estimated at 20 per cent of the building heat transmission loss, but in church vestibules and similar places where the openings are large and much heat used in proportion to the size of the space, the leakage may be equivalent to 100 per cent of the building heat loss. The method provides easy means of checking actual results against the estimated load, but cannot be successfully applied without judgment, backed by experience. Checking the estimated load in the Butler Field house, which was a tight building with an insulated roof, against the test results, proved that 20 per cent allowance for leakage was substantially correct.

Leakage is the unpredictable item in a corrugated iron building of the type described. The necessary height of the building, in order to provide the fall for the coal screens, etc., produces a heavy chimney effect. The construction necessarily leaves many holes. Where a corrugated sheet is bolted to an angle, there is a hole the size and shape of the corrugation every  $2\frac{1}{2}$  in. around the entire building, both top and bottom. Where loading chutes pass through the

steel floor or the sidewall of the track extension, the chutes themselves form tubes which admit cold air to the building. Effective ways of closing them are difficult to devise. Also, there is an inspection opening in the chutes just above the steel floor that is open a good deal of the time. In this particular case, there is a large water settling cone in which the water is reclaimed and sludge drained off by the settling process. The bottom of this cone is enclosed in a housing, but the top is exposed and few precautions are taken to make it tight where the housing fits around the cone. Also, an additional part of the load is that required to keep the water in the cone from freezing and this is accomplished by the simple fact that its lower half is housed inside the building.

There are various other openings and holes through which air can pass. It is difficult for a person to realize without an inspection how open these buildings are. It is equally difficult to arrive at a conclusion as to how much to figure for the infiltration load. In this case, arriving at the infiltration load was purely guess work, and it was guessed at 100 per cent of the measurable quantities.

Based upon the foregoing estimated information, a furnace rated to deliver 4,000,000 Btu per hour was selected. The basis of the rating was that given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, limiting the rating to 3500 Btu per square foot of surface. Fig. 2 shows a shop photograph of the furnace containing 1145 sq ft of heating surface. It was equipped with a special type of economizer designed to fit the limited space available for the location of the plant. The ratio of grate area to heating surface, if grates were provided for a furnace of this size, would be about 1 sq ft of grate area to 47 sq ft of heating surface. This ratio, combined with the low temperature air being driven over the heating surface, would result in a very high efficiency, and should compare with the test results on the Butler Field House, where under somewhat similar conditions of large volumes of air handled, efficiency was shown at 89.4 per cent.

The furnace was equipped with a 400 lb stoker which, together with the efficiency of the furnace and the type of coal to be used, could reasonably be expected to produce a 4,000,000 Btu output.

The blower had a 54 in. x 54 in. wheel, was equipped with a 10 hp motor driving the fan at about 160 rpm, and delivered approximately 47,000 cfm. This would handle the volumetric contents of the building about every 7 min, and it would result in a temperature rise of the air as it passed over the furnace at about 80 deg, which was about as low as would be feasible under the extreme conditions of a building of this character.

Fig. 3 shows a floor plan of the heating equipment and Fig. 4 is a floor plan and elevation of the building and the location of the plant with relation

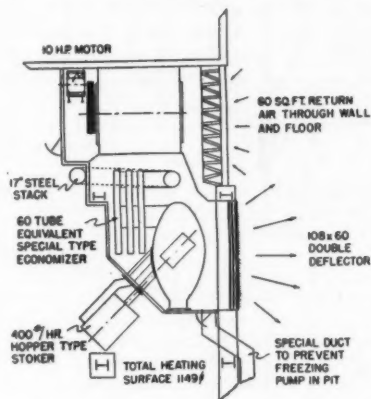


FIG. 3. LOCATION OF HEATING EQUIPMENT

to the building. It will be noticed that the heat was all discharged at one place, and the return air all brought back to one point. There was no short circuiting as the discharged air was free to move away, with considerable velocity, while the return air flowed back on a lower level.

The system was originally designed without any distributing ducts, which would be objectionable for obvious reasons. But when 17 deg below zero

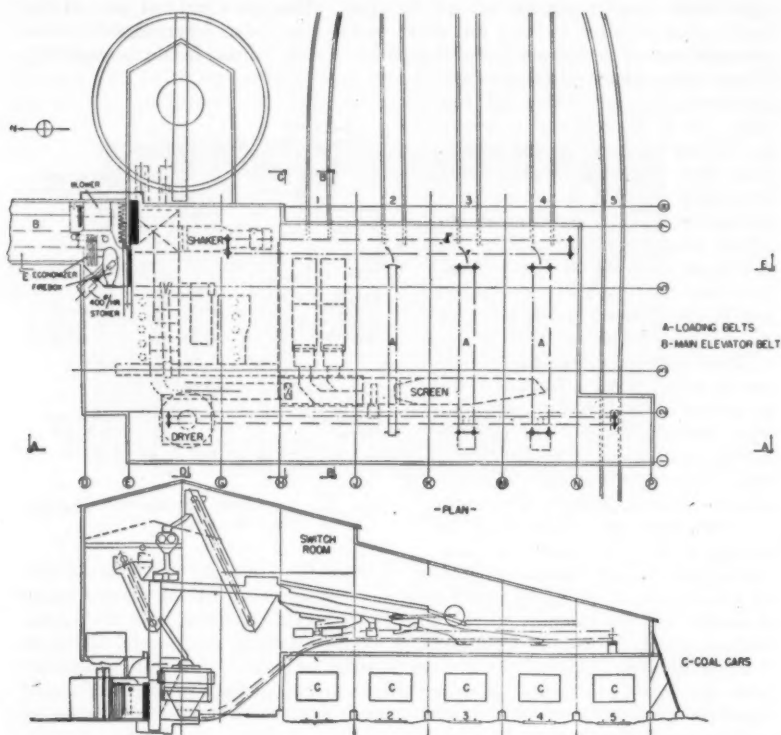


FIG. 4. FLOOR PLAN AND ELEVATION SHOWING ARRANGEMENT OF HEATING PLANT FOR COAL TIPPLE

weather was encountered one pump froze, due to the fact that it was set in a pit on a lower level than the return air inlet and no provision had been made to draw the air out of the pit. To cure that particular trouble, a special duct was run, discharging its heat downward immediately over the pump, thereby combining spot heating with the attempt to heat the space as a whole.

Also, on account of the excessive leakage, the temperature differential between the bottom and top of the tippie was quite high. At 17 deg below zero outside, the temperature was in the neighborhood of 65 F in the upper part of the tippie, and down on the level of the return air, approximately 40 F.

The cure for this condition would consist in increasing both the amount of heat and the volume of air, and in making the building tighter to prevent the excessive infiltration of large volumes of below zero air.

The net result of this experimental installation with the adjustments that have been made and with some tightening up still to be done, is satisfactory to the owners, because it enables them to get away from troubles which they had experienced before when using spot heating with steam and with unit heaters. They were not interested in fuel economy. They built a conveyor which takes the coal right from the washer and dumps it into the hopper of the stoker. They were interested in eliminating freezing of pipes, pumps, etc., and getting rid of the action of sulphur water which prevails around so many mines. As a part of the portable building, this plant can be picked up with proportionately as little trouble as the balance of the building and reset in another location. For these reasons, the purchasers feel that the experiment has been a success.

## DISCUSSION

W. H. CARRIER, Syracuse, N. Y.: The author had a very difficult problem but has done an excellent job of *guesstimating*. In the early state of the art of calculating warm air heating for buildings as I knew it, back in 1901 and '02, all of our buildings with fan system heating had to be *guesstimated*.

My first experience was with the Buffalo Forge Co., when I worked in the drafting room and the estimating department. The method which I found then in practical use was rather appalling to an engineer. It was a system of *guesstimating* in the truest sense of the word. Fan systems, or hot blast systems as they were then called, were usually employed for large spaces, such as, factories and store-rooms. An entirely *rule of thumb* system was generally employed and as I recall it, it went about this way:

If the building were of average construction in respect to glass and walls, openings, etc., the air per minute supplied by the fan was taken at approximately one-twentieth of the cubic feet contents, that is, a 20-minute air change. It had been found by experience that for climates similar to that of New York State or Pennsylvania, 5 sections deep of 4-row 1 in. pipe coils were sufficient in most cases. In the South, 4 sections of coils would be employed and in more northern climates, such as Canada and northwestern United States, 6 sections of coils would be employed.

If the building had more than 30 per cent glass or if it were unusually leaky, then we played safe and used a fifteen-minute air change. On the other hand, in a building with small glass area and of unusually good construction, we would go to a thirty-minute air change. Following this procedure, obviously many jobs had an excess of heating capacity and a certain number would fail to meet the guarantees. The firm that took the most chances would have the lowest price and usually got the contract. This, of course, led to a good deal of chiseling in estimating and more failures than were desirable. At this time, there was no exact knowledge of the heating capacity of coils at different air velocities, different steam pressures, etc., and even if we had had an exact method of estimating heat losses in Btu, we had no data by which to select the equipment to balance these heat losses.

After I had been in the estimating department for six months, I thought I saw a means of getting the company more business by closer estimating and at the same time safeguarding the job by a reasonable margin. Data were available, largely as a result of the work of Peclet which was used in Carpenter's book on heating and ventilating, for obtaining a reasonable estimate of building heat losses. These only required an additional margin to allow for infiltration to estimate the heat requirements of the building. Therefore, I proposed to the management that I be permitted to make tests of the heaters in various depths and at various velocities and tempera-

tures in order to determine the heating capacity of the apparatus which then was not definitely known. This was done early in 1902 and the results were worked up in a comprehensive set of heater tables practically identical with those that are in use today by various companies manufacturing fan system air heaters. These, I believe, were the first data of the kind ever obtained. They preceded the tests and tables for vented cast-iron heaters by about three years. With these data in hand, we no longer figured definite air changes but started with a definite depth of heater and supplied the air quantity necessary to heat the building. This did two things. First, it insured a minimum combined cost of fan, dustwork and heater and second, it made practically every job installed fulfill the guarantee.

In the winter of 1902 and 1903, tests of existing installations were made to determine the actual heat balances as compared with the revised calculations. From these, the allowances which should be made for infiltration were determined. Carpenter had recommended one hour infiltration as a rule and up to one-half hour for small buildings with exceptional exposure. The actual infiltration changes were found, on large buildings, to be much lower than this. The minimum infiltration we found in heating a large car repair shop was once in four hours. We still had to *guesstimate* on the infiltration air change. However, we included our factor of safety and compromised on an infiltration air change of between one and two hours, depending on the size and construction of the building. This proved universally adequate.

Beginning in 1902, all installations made by our firm were made on a Btu basis. This was the history of the early progress in the art and of the development of more rational methods of calculating heat losses. When you come back to a proposition such as Mr. Campbell's, you still have to go back pretty well to the old method of *guesstimating*.

C. M. ASHLEY, Syracuse, N. Y.: We have two research programs covering the subject of infiltration; one of these having to do with techniques of measurement, which still can be very much improved. The other has to do with infiltration through doors and openings. I believe that infiltration must be recognized as still one of the least known subjects in the whole art.

B. B. REILLY, Pittsburgh, Pa.: I would like to question the author's basis for design. Was it for comfort only, or were other factors to be considered; 3,600,000 Btu per hour seems like a rather extravagant use of heat for this type of building which is normally very lightly occupied—a half dozen people being the ordinary population. We have had satisfactory success with spot heating for comfort in some of the eastern coal preparation plants using much lower heating requirements.

**AUTHOR'S CLOSURE:** If the comfort of the employees was the principal consideration, that is correct; it would have been a lot of heat. The principal consideration was to prevent shutdowns due to freezing. That has been the bane of tippie operation, particularly in these large coal-washing plants in connection with strip mines, where the shovels pick up a lot of mud along with the coal. The only way it can be made fit to put on the market is to wash the mud out, and hence it is a matter of preventing freezing, which would shut down the operation of the tippie entirely, if permitted.

The remarks made by Dr. Carrier indicate that, while we do like to have things scientifically accurate, and we are making great progress in that direction, the contracting engineer particularly is bound to come face to face with problems in which he has to base his decisions, his design, on general experience and sometimes thumb rules.

We have a long way to go before we shall be able to solve all these problems scientifically. Additional experience, not only with this plant, but with two others in different locations, has confirmed the rule established by the author on the first one, namely, that it is fairly safe to calculate the heat load by taking the total of the transmission losses plus one hundred per cent for leakage. Even then, there will have to be reasonable care in stopping up holes and making the building comparatively tight.



**1257**

## THE ENGINEERING CONTROL OF SOME SOLVENT HAZARDS IN WAR INDUSTRIES †

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### INTRODUCTION

THE ENGINEERING control of solvent hazards in War Industries, however magnified or spectacular they may appear, due to increased production demands, follow in principle the same sound, practical, common sense, proven applications which were developed as an outgrowth of peace time initiative, foresight and progressiveness. New problems and intensification of old ones serve only to challenge the imagination of those directly concerned with this vital problem.

The broad, over-all aspects of control and prevention of ill health resulting from the careless, ignorant misuse of industrial solvents can best be effected by mutually integrating the cooperative efforts of management, the worker, the community at large, industrial physicians, nurses, safety men and industrial hygiene engineers. In the present discussion an attempt is made to outline the role of the industrial hygiene engineer in the great drama unfolding itself when our very existence as a free nation depends so much upon the output of our war industries.

To emphasize the importance of engineering control, some of the apparent toxic effects of chemicals, especially solvents, will be reviewed.

According to Foulger,<sup>1, 2, 3</sup> the initial reactions of living, intact human and animal organisms to foreign chemicals, which when introduced into the body may have a systemic effect (as opposed to a purely local action), are the same regardless of the structure or physical properties of the chemical. They are the same no matter what the route of absorption of the material. It seems quite possible that serious organic injury from harmful chemicals occurs more often as a result of an acute exposure of relatively short duration, superimposed upon the accumulated effects of a prolonged low grade exposure, than it does from individual, well isolated, acute incidents without the background of chronicity.

To quote Foulger again, "First effects of exposure to toxic chemicals consist of a few simple symptoms and certain definite signs, which are the same in exposures of many kinds. The symptoms are: easiness of fatigue, headache, gastro-enteric disturbance (nausea, loss of appetite, a feeling of fullness of the stomach, gas on the stomach, pain in epigastrium), dizziness, precordial pain, pain or tingling in the extremities, and dyspnea on slight exertion. Of course, not all are present in all cases. They are, at first, indi-

† Similar information was given in address at the Greater New York Safety Conference 30, March, 1944.

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<sup>1</sup> Superior numerals refer to Bibliography.

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cations of functional disturbance only and not of organic injury. Of these symptoms, easiness of fatigue is probably the most universal and usually the first to appear."

Von Oettingen<sup>4</sup> states: "In the paraffin series the first fractions (pentane and hexane) are relatively non-toxic, having low narcotic and irritating properties. The next higher boiling fractions have more marked narcotic and irritant properties and a narrow margin of safety (difference between the minimum narcotic and minimum lethal concentrations). The lowest boiling fraction of the olefines also has narcotic properties but a comparatively wide

TABLE 1—PROPERTIES OF SOME IMPORTANT

SOLVENT	COMMERCIAL SYNONYM	LIMITS OF INFL. BY VOL. PER CENT		EXPLOSIVE RANGE	SPECIFIC GRAVITY	
		Lower	Upper		Sp. Gr.	Comp. with Air
Acetone.....	Dimethyl ketone...	2.5	12.8	10.3	0.790	2.0
Amyl acetate.....	Banana oil.....	1.1		(F.P.-94 F)	0.868	4.5
Aniline.....	Aniline oil.....			(F.P.-131 F)	1.022	3.22
Benzene.....	Benzol.....	1.4	6.8	5.4	0.879	2.7
Benzine.....	Petroleum vapors...	1.1		(F.P.-0 F)	0.625	4.48
Butanol.....	Butyl alcohol.....	1.7		(F.P.-115 F)	0.811	2.55
Butyl acetate.....	n. Butyl acetate.....	1.7		(F.P.-100 F)	0.883	4.0
Carbon tetrachloride	Tetrachlormethane.	(Non-Inflam.)			1.60	5.3
Chlorobenzene.....	Monochlorobenzene			(F.P.-81 F)	1.106	3.8
Ethanol.....	Alcohol.....	2.3	19.0	16.7	0.789	1.6
Ethyl ether.....	Ether.....	2.3	6.15	3.85	0.708	2.55
Ethylene dichloride.	Dutch liquid.....	6.2	15.9	9.7	1.25	3.42
Gasoline.....		1.5	6.0	4.5	0.747	3.4
Methanol.....	Wood alcohol.....	6.7	36.5	29.8	0.793	1.1
Naphtha.....	Stoddard solvent.....	1.2	6.0	4.8	0.788	4.9
Perchloroethylene...	Tetrachloroethylene.	(Non-Inflam.)			1.608	5.72
S-Tetrachlorethane.	Acetylene Tetra- chloride.....	(Non-Inflam.)			1.60	
Trichloroethylene...	Westrosol.....	{ Practically Non-Inflam. }			1.46	4.5
Toluene.....	Toluol.....	1.3	6.8	5.5	0.866	3.2
Turpentine.....	Spirits of turpentine	0.8		(F.P.-95 F)	0.860	
Xylene.....	Dimethylbenzene...	1.0	6.0	5.0	0.88	3.68

(NOTE—Recommended by following states: (1) Cal., Colo.; (2) Cal., Colo., Kan., Ky., Mass., Minn.; (6) Cal., Colo., Kan., Ky., Mass., Minn., Okla., Penna., Wis.; (7) Cal., Colo., Conn., Kan., Ky., Md., Wis.; (10) Cal., Colo., Kan., Mass., Mich., Minn., Okla., Wis.; (11) Mich.; (12) Mich.; (13) Cal., Colo.; (16) Mich.; (17) Kan., Mass., Mich., Minn., Okla., Wis.; (19) Cal., Colo., Conn., Kan., Ky., Md., Mass.,

margin of safety, while the higher homologues are more potent as narcotics and are more irritating. The lower boiling fraction of cycloparaffines has narcotic properties but is less irritating than the corresponding olefines. The higher boiling cycloparaffines and the unsaturated paraffines are potent narcotics and have a narrow margin of safety.

"The lowest boiling fraction of aromatic hydrocarbons (benzene) has narcotic properties and produces injurious effects on the blood and blood forming organs. The next two higher boiling fractions (largely toluene, ethyl benzene and xylene) have slightly greater narcotic and irritant properties and a smaller margin of safety. They have, however, less marked hematotoxic action. The

highest boiling fraction has marked narcotic properties but is least dangerous because of the low volatility."

"The main difficulty encountered in determining the health hazards of hydrocarbon solvents arises from the fact that different brands of solvents have varying percentage composition of paraffines, olefines, cycloparaffines and aromatic hydrocarbons."

Flury<sup>6</sup> classifies the toxicity of common solvents as follows:

1. Those which are harmless under ordinary conditions of industrial use, but dangerous if their vapors are breathed in high concentrations.

#### SOLVENTS USED IN WAR INDUSTRIES

B. P.	TOXIC LIMIT* PARTS PER MILLION	HAZARDS			EFFECT ON MAN
		Fire	Explosion	Dermatitis	
56° C	200 (1)	+	+	+	Irritant, dizziness
147.6	400 (2)	+		+	Irritation, dizziness, nausea
184.4	5 (3)				Cyanose, excitation, coma
80.0	100 (4)	+	+	+	Acute narcotic, chronic bleedings, fatigue, blood changes
35-80	1000 (12)	+	+	+	Intoxication, narcotic
117.7	100 (5)	+		+	Excitation
126.5	400 (6)	+		+	Irritates eyes, nose, throat
76.8	100 (7)			+	Irritates nose, eyes, throat; numbness, liver damage
132.0	75 (14)	+	+	+	Acute narcotic, chronic headache, giddiness
78.0	250 (8)	+		+	Excitation, intoxication
34.4	400 (9)	+	+	+	Anesthetic, irritant, narcotic
83.5	100 (10)			+	Dizziness, nausea, narcosis, giddiness
50-140	1000 (19)	+	+	+	Intoxication, narcotic
64.0	100 (20)	+		+	Nerve poison, irritant, dizziness, blindness
156.0	5000 (21)	+	+	+	Intoxication, narcotic
120-1	100 (11)			+	Similar to trichlorethylene, but less toxic
146	10 (13)			+	Narcotic, liver damage, irritation, headache
86.7	100 (16)	Slight		+	Nervous disturbances, dizziness
111.0	100 (15)	+			Fatigue, dizziness, slight blood changes
116.7	200 (17)	+		+	Irritation, headache, renal disturbances
139.0	100 (18)	+	+		Fatigue, dizziness, slight blood changes

(3) Cal., Colo., Kan., Mass., Minn., Okla.; (4) Cal., Colo., Conn., Penna., S. C.; (5) Cal., Colo., Kan., Okla.; Mass., Minn., Okla., Penna., S. C., Wis.; (6) Cal., Colo.; (7) Cal., Colo., Kan., Mass., Mich., Minn., Okla., Kan., Ky., Mass., Minn., Okla., Wis.; (8) Cal., Colo.; (9) Cal., Colo., Kan., Mass., Mich., Minn., Okla., Kan., Ky., Mass., Minn., Okla., Wis.; (10) Cal., Colo., Kan., Mass., Minn., Okla., Wis.; (11) Cal., Colo.; (12) Cal., Colo., Conn., Ky., Md., S. C.; (13) Cal., Colo.

2. Those which cause secondary reaction in the body, from which a quick recovery is possible if the damage is not severe.

3. Solvents which may cause secondary effect which, when not fatal, may be irreparable. This group must be handled with the greatest of care under proper controls.

Time does not permit running the entire gamut of solvent hazards. Consideration, therefore, is devoted to a few of the most common types which are finding widespread use in war industries. Physical, chemical and toxic properties are summarized in Table 1. The generally recognized hazards in the use of organic solvents are fire and explosion, bodily contact, and systemic

toxicity. In this paper, consideration will be given only to the possible control of toxic effects.

#### ENGINEERING CONTROL

The engineering control of atmospheric contamination due to the misuse of toxic volatile solvents is primarily one of the most important functions of the industrial hygiene engineer and may best be achieved by one or more of the following general methods: (1) substitution of less toxic materials, (2) isolation of those processes which produce contamination, (3) dilution with uncontaminated air, (4) control at point of generation or dissemination, (5) respiratory protective devices, and (6) maintenance, housekeeping, and education of the worker. To illustrate the specific application of these prin-

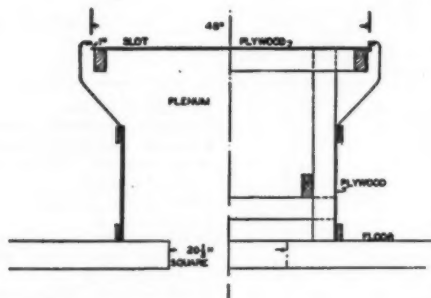


FIG. 1. SECTION ON LATERAL VENTILATED TABLE. AIR FLOW = 50 CFM PER SQUARE FOOT TABLE SURFACE

ciples, a few important solvents have been selected and an attempt will be made to show by their typical usage how best their attendant hazards may be minimized in war industries.

**Benzene** (Benzol— $C_6H_6$ ): Benzene, an extremely volatile and inflammable solvent, is used as a basic material for the manufacture of aniline, picric acid and phenol. Under various trade names it is used as a substitute for toluol in paint removers. As a solvent in the coating and cementing of fabrics with natural rubber and other operations, benzene is utilized extensively. It is probably the best natural solvent or softening agent with or without the addition of some chlorinated hydrocarbons or a ketone, for example, butanone. The cementing hazard referred to previously has been most effectively controlled by employing cements containing toluol or trade name processed petroleum which contain a large proportion of toluol and by conducting the cementing on lateral or downdraft ventilated tables. Design sketches (Figs. 1 and 2) of these tables will be found in a paper published on this subject by Thomas and Tebbens.<sup>6</sup>

A solvent naphtha such as hydrogenated naphtha has been used successfully in place of benzene as a flotation agent for the polishing material in the polishing of reflectors.

Inasmuch as the nitration of benzene is an enclosed process, the small amount of benzene liberated into the general room atmosphere has been kept at safe

levels by means of: (1) roof ventilators with mechanically driven fans, and (2) doors and windows opened as much as possible. Workers making repairs on benzene nitrators are required to wear gas masks (chemical-filter respirators) or U. S. Bureau of Mines approved (Type B Hose Masks) supplied-air respirators. Choice depends on the type of repair and duration of exposure.

Wilson,<sup>7</sup> as a result of his studies of 1,104 workers engaged in the manufacture of synthetic rubber, suggests keeping benzene concentration well below 100 ppm.

In the one noted British case<sup>8</sup> of chronic benzene poisoning which ended fatally in 1942, anemia was attributed to exposure to rubber solvent said to contain not more than 5 per cent of benzene. Estimation of the benzene or of a mixture of benzene and toluene in the air in the vicinity revealed 1 part in

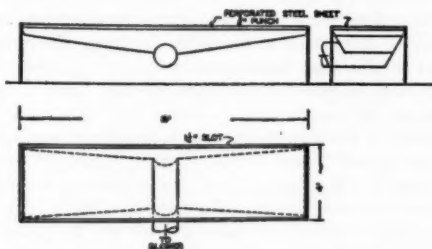


FIG 2. DOWNDRAFT VENTILATED TABLE

10,000 (100 ppm). One non-fatal British case arose as a result of the use of airplane dope containing less than 15 per cent benzene in the mixture.

According to Henry,<sup>9</sup> chronic benzene poisoning in England is less common, probably because of increased precaution.

**Toluene** ( $C_6H_5CH_3$ —phenyl methane, toluol, methyl benzene): Toluene is used in the manufacture of explosives, drugs, perfumes and dyes. Widespread use is made of it as a solvent for gums, resins, oils, and many types of cellulose esters. It is used in the lacquer coating for impregnating fabrics, paper and articles made of various other materials. In the aircraft industry it is used as a thinner in a special paint necessary for coating fusilages, wings, etc., where application is by dipping and spraying. Due to the acute shortage of toluene, caused by its being diverted to explosive manufacture, several possible substitutes have been used by lacquer manufacturers and users. A petroleum product called hydrogenated naphtha, the aliphatic content of which is much higher in the aliphatic series than the straight run petroleum naphthas, offers an interesting possibility. For most uses, such new mixtures are better solvents than the straight run naphthas. They have a surprisingly high degree of tolerance or compatibility of nitrocellulose in lacquers and can be substituted for toluene almost gallon for gallon in such formulation. The use of hydrogenated naphthas decreases the health hazard as well as the fire hazard.

Based upon a study of 106 painters in a large airplane factory exposed to the inhalation of toluene from 100 to 1100 ppm for periods ranging from two

weeks to more than five years, Greenburg et al.<sup>10,11</sup> found that: (1) Exposure of human beings to toluene resulted in enlargement of the liver in 30.2 per cent, perforated nasal septum (probably due to zinc chromate) in 4.7 per cent of the men, (2) erythrocyte counts were low, 17 per cent below 4.5 million erythrocytes per cubic millimeter of blood, (3) hemoglobins were high, having 16 gm or more per 100 cc of blood, (4) absolute lymphocyte counts high, although differentials were normal, and (5) mean corpuscular volume was high.

Von Oettingen et al.<sup>12</sup> suggests a maximum permissible concentration for an 8 hour day as 200 ppm for toluene. In operations where specific accident hazards exist this may be too high.

As a control in the nitration of toluene, the nitrators are enclosed and connected to an acid recovery plant by means of a local exhaust ventilation system. Nitrators are maintained under negative pressure sufficient to prevent the escape of toxic vapors into the room atmosphere during the nitration cycle.

**Carbon Tetrachloride** ( $\text{C Cl}_4$ ) Tetrachloromethane: Carbon tetrachloride is a heavy, volatile incombustible solvent and extractant for fats and oils. Because of its incombustibility it is used extensively in drug, chemical, rubber, paint, rubber cement and textile soap manufacture.

Twenty-five per cent carbon tetrachloride is used with a cutting compound in the critical tapping and machining operation of a gun plant. As an efficient substitute, 8 per cent of the less toxic, trichlorethylene was substituted.

According to the United States Public Health Service,<sup>13</sup> the cause of illness among 135 employees in a Kentucky plant manufacturing parachutes was traced to carbon tetrachloride used in the cleaning of soiled spots on the *chutes*. The first symptoms noted were coincidental with the beginning of the heating season and consequent reduction of ventilation in an effort to conserve fuel. The cleaning of these nylon parachutes is now being done with mild soap and water.

Solvent vapors of carbon tetrachloride are given off when tracer and igniter mixtures are charged into the copper jackets in explosive manufacturing. As precautionary measures the following are effective: (1) sufficient general ventilation be provided, and (2) charging machines and dies be cleaned with either trichlorethylene, ethyl alcohol or coal oil instead of carbon tetrachloride.

In the centrifugal removal of carbon tetrachloride from degreased objects, control is best effected by enclosing the centrifuge and exhausting the enclosure at a minimum of 100 cfm per square foot of enclosure opening.

Based upon 11 typical plants investigated, Elkins<sup>14</sup> believes the toxic limit of 100 ppm for carbon tetrachloride is too high. He suggests 25 to 50 ppm.

Smyth, et al.,<sup>15</sup> state that 50 to 80 ppm of carbon tetrachloride are detectable by odor by the average individual.

When carbon tetrachloride is used openly, as in a shop, ample ventilation should be provided and approved respiratory protection be furnished each man. Skin contact should be avoided.

**Gasoline:** Gasoline, because of its convenience and traditional usage, is used extensively for washing hands and arms. However, sufficient precautions are not taken to see that the gasolines used do not contain tetraethyl lead. Even as a good substitute for ordinary gasoline for washing and cleaning purposes, mineral seal oil, mineral spirit or various grades of naphtha may be recommended.

According to Humperdinck,<sup>16</sup> German gasoline contains an average of 0.8 cc of ethyl fluid per liter. The fluid is composed of 63.0 per cent lead tetraethyl,

25.8 per cent ethylene dibromide, 8.7 per cent ethylene dichloride and 2.5 per cent coloring matter. Volatility is sufficient at normal temperatures to produce a concentration of 5 milligrams of lead per liter of air.

Serious exposures to lead are encountered in the spray cleaning with leaded gasoline of airplane motors after they have been tested. The use of tetraethyl gasoline for this purpose should be prohibited. Spray cleaning should be done in a small room, cabinet or booth provided with mechanical exhaust ventilation rate at a minimum of 150 cfm per square foot of room

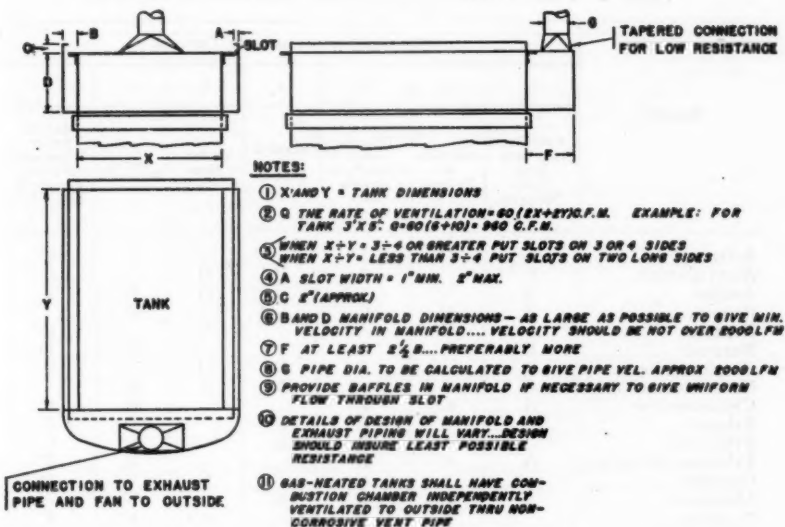


FIG. 3. VENTILATION OF DEGREASING TANK

cross-sectional area or booth opening. As an added precaution the operator should remain always in the clean air upstream of the motor.

**Degreasing Solvents:** The chlorinated aliphatic compounds, because of their non-inflammable nature and strong solvent action, are the most widespread solvents used in the degreasing of metals. The two principal ones today finding extensive use for this purpose are trichlorethylene (Perma-a-chlor, Triad, Tromex, Blacosolv, Tri-chlor), and perchlorethylene (Phillsolv, tetrachlorethylene, Per-chlor).

Degreasing is usually accomplished by the dipping, wiping, vapor, vapor spray, hot liquid-vapor or hot liquid-cold-liquid vapor processes.

Based upon an engineering study of 108 degreasers, Morse and Goldberg<sup>17</sup> state that degreasing liquid cleans metal parts by (1) immersion, (2) spraying, or (3) vapor. Ninety per cent of all degreasers use two or more methods. According to the authors, degreasers usually consist of a tank with a heating unit and condensing jacket, but in a recent development where the higher

boiling tetrachlorethylene is used, there is no condensing jacket and the temperature is controlled thermostatically.

Average atmospheric concentrations found were: 96 ppm for ventilated condenser machines; 135 ppm for non-ventilated condenser machines and 221 ppm for non-condenser type machine. They attribute excessive concentrations to the following: 1) speed with which work is lowered into and removed out of the machine (manufacturers recommend 12 fpm), (2) cool-

TABLE 2—TOXIC CONTROL BY STANDARD ENGINEERING METHODS

SOLVENT	ENGINEERING METHODS OF CONTROL						
	1	2	3	4	5	6	7
	Substitution with Less Toxic Material	Isolation of Process	Dilution with Uncontaminated Air	Respiratory Protection	Protective Clothing	Local Exhaust	Maintenance, Housekeeping and Employee Enlightenment
Acetone.....	E <sup>a</sup>	D	A	...	...	B	C
Amyl acetate.....	E	...	A	B	D	...	...
Aniline.....	A	...	...	...	D	B	E
Benzene.....	A	D	...	...	...	B	...
Benzine.....	...	B	A	...	D	...	C
Butanol.....	E	B	C	...	...	A	D
Butyl acetate.....	...	...	A	B	D	...	C
Carbon tetrachloride	A	C	...	D	E	B	...
Chlorobenzene.....	A	C	D	...	...	B	E
Ethanol.....	...	B	A	E	...	D	C
Ethyl ether.....	...	A	B	E	...	D	C
Ethylene dichloride.	A	C	D	...	...	B	E
Gasoline.....	...	A	C	...	E	B	D
Methanol.....	A	C	D	E	...	B	...
Naphtha.....	...	E	B	...	A	D	C
Perchloroethylene...	...	B	A	...	...	C	D
Tetrachlorethane...	A	C	...	E	...	B	D
Trichloroethylene...	E	A	C	...	...	B	D
Toluene.....	B	D	C	E	...	A	...
Turpentine.....	...	E	A	D	...	B	C
Xylene.....	...	C	A	D	...	B	E

<sup>a</sup> NOTE—Choice of Method: "A," First choice; "B," Second choice; "C," Third choice; "D," Fourth choice; "E," Fifth choice.

ing beyond the dew-point temperature of the room, causing condensation and addition of water to the solvent, thereby lowering its boiling point, (3) not keeping work in the vapor zone until condensation stops, (4) poor arrangement of work in basket, (5) lack of temperature control of circulating water, which should be below 110 F, and above all (6) operation by inexperienced and unintelligent workers.

Morse and Goldberg recommend: Mechanical hoists set at a maximal speed of 20 fpm and withdrawal of work at same speed; equipment with thermostatic control of heating and condensing zones; proper design of basket; location of tanks in as large an area as possible, maintaining a minimum

operative area exposed to air, a minimum freeboard distance of 0.6 of the tank width, careful maintenance of heat balance with every change in type and weight of metal, and prevention of dead air spaces in work.

Solvent vapor concentrations are usually lower when tanks are small (less than 10 sq ft in cross-sectional area), well located in large rooms with high ceilings, and not near open windows or doors. However, if the general room ventilation is poor, tanks are large, or located near open windows or doors

TABLE 3—TYPES OF CANISTER FOR PROTECTION UNDER VARIOUS CONDITIONS  
(Gas Masks)

CONDITIONS	CONTENTS OF CANISTER	COLOR OF CANISTER
(1) Protection against organic vapors, such as aniline, gasoline, benzene, ether, toluene, and the like (when not over 20 per cent in air).	600 cu milliliters or more of activated charcoal.	Black
(2) Protection against acids such as hydrochloric, sulphur dioxide, nitrogen peroxide, chlorine and the like (not over 1 per cent in air).	600 cu milliliters or more of soda lime or fused caustic soda.	White
(3) Protection against ammonia gas not over 3 per cent. The 3 per cent limit cannot be long endured by the wearer because of the skin irritation from the gas.	Copper sulfate and charcoal.	Green
(4) Protection against carbon monoxide (not over 3 per cent).	A mixture of metallic oxides known as <i>hopcalite</i> which catalyzes the combustion of carbon monoxide with oxygen from the air and produces carbon dioxide.	Blue
(5) Protection against all of the above gases; the <i>all service mask</i> .	All of the absorbents mentioned above, but only small amounts of each.	Red
(6) Protection against a combination of organic fumes and acid fumes.	Activated charcoal and soda lime.	Yellow
(7) Protection against ammonia and smoke.	Copper sulfate and charcoal with a filter pad for smoke.	Brown
(8) Protection against hydrocyanic acid gas (not over 2 per cent).	Caustic soda impregnated on pads.	With a green stripe.

or in small rooms, local exhaust should be provided by means of slot-type, lateral exhaust hoods located along one or both long sides of the tank at the upper edge (see Fig. 3). The minimum effective exhaust ventilation rate is computed as follows:  $Q = 50 LW$ ; where  $Q$  is the exhaust ventilation rate in cubic feet per minute;  $L$  is the length of the tank in feet; and  $W$  is the width of the tank in feet. Remedial measures are indicated: if the atmospheric concentration of trichlorethylene is more than 200 ppm; if the solvent consumption is more than 2 gal per square foot of tank area per 100 lb of operation, or if the odor of trichlorethylene is distinctly noticeable.

Small amounts of aniline have recently been added to commercial degreasing solvents as a substitute for the triethylamine which serves as a stabilizer. In view of the high toxicity of aniline, greater care must be exercised.

In some cases attributed to solvent poisoning recently appearing in the literature,<sup>18</sup> the workers so affected were engaged in cleaning a degreasing tank without proper breathing apparatus or life belts.

An unusual trichlorethylene fatality reported in England<sup>19</sup> involved a night watchman in Birmingham, who tried to dry-clean his trousers by dipping them in industrial trichlorethylene. After taking the trousers into a shelter with him, turning on an electric radiator and going to sleep, he was killed by the vapor.

Reference again to Table 1 shows a few additional, very interesting solvents which the Army is called upon to find practical means of controlling. Fortunately the backlog of valuable information accumulated in the past on how to keep workers well is a most treasured non-secret weapon available with which to do the job at hand. Generally speaking, from an engineering point of view, it might be added with a measured amount of confidence that few, if any, of the problems encountered have failed to yield their hazardous status to good, sound, practical, common sense applications.

#### SELECTION OF TOXIC CONTROL METHOD

Table 2 will be useful in selecting a toxic control procedure. Various methods are shown in the order of their preference and desirability.

When gas masks are used for respiratory protection, canisters must contain absorbents for the toxic gases encountered. Standard markings for gas mask canisters which indicate their contents and protective ability are shown in Table 3.

#### CONCLUSIONS

In closing this discussion there are three points that cannot be over-emphasized:

(1) Industrial solvents, in general, are capable of entering the human body by skin absorption as well as by mouth or inhalation. In attempting control, therefore, all these vital possibilities should be considered by the engineer.

(2) In addition to knowing the physical, chemical, corrosive or explosive properties, the engineer must familiarize himself with the *chronic* as well as acute toxic results anticipated from ignorant or excessive use of these substances.

(3) All solvents, regardless of their toxicity, unless proven otherwise, irrespective of their mode of entry into the human organism, are foreign to the normal body metabolism and thus contribute no end to the accident proneness of the individual.

With the foregoing thoughts borne uppermost in the mind of the engineer, the solution of one of the major problems directly affecting the conservation of manpower in war industries will be relatively simple. "Death Without Battle," to borrow the title from a recent Army educational "skit," insofar as applied to solvent hazards, will be an expression relegated to the Sargasso Sea of obsolescence.

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#### DISCUSSION

L. T. AVERY, Cleveland, Ohio: Until recently it has been sufficient to consider that if we handled air, if we pushed it and pulled it and filtered and cooled and heated it, that was all we had to do. We are now asked our opinions as to what quality of air is required for certain purposes. This paper is of value to us as heating and ventilating engineers; although it is possibly old material to an industrial hygienist. For example Jack & Heintz built a new building, in which there was a bearing assembly room. A fine air conditioning system was installed that was a failure. It was a failure because the emphasis as far as the management and the architect were concerned was placed on temperature and humidity. The system

was designed for 78 deg dry-bulb, 40 per cent relative humidity, chiefly to keep the bearings from tarnishing. There were 200 girls working in the room. They used as solvents carbon tetrachloride, gasoline and naphtha. Now, if you look at Table 1 you will see that a concentration of 100 parts per million of carbon tetrachloride is toxic, it irritates the nose, eyes and throat and causes liver damage. A thousand parts per million of gasoline is a fire explosion hazard and causes dermatitis. Somebody thought the carbon tetrachloride mixture with the gasoline maybe reduced the fire and explosion hazard. Naphtha at 5,000 parts per million is toxic, it is a fire hazard, an explosion hazard, and dermatitic in its action.

The air conditioning system was set up on the basis of 17,000 cfm, 12,000 cfm of which was *recirculated air*.

In reference to Table 2, the recommendation for handling carbon tetrachloride, the preferred recommendation, is substitution with less toxic material. For handling gasoline it is isolation of the process. For naphtha, it is protective clothing.

Well, this was an assembly operation with 200 girls sending out thousands of these bearing assemblies, and the first solution that we tried was to follow column 3, dilution with uncontaminated air. We could do that in the spring, but this 40 per cent relative humidity became important as soon as we got into warm weather. The question was then: Shall we put in additional cooling and dehumidifying to reduce 12,000 cfm outside air to this design condition, or shall we treat the recirculated air? In that particular case activated carbon was used on the recirculated air at a rate of 15 cfm per canister. This cleared the employee bad health reactions, made the refrigeration system effective and turned a failure into a successful system.

I bring this to your attention because if anybody talks to you about installing air-conditioning in a manufacturing process it is your business to inquire whether solvents are to be used.

One other problem of an entirely different nature came when we designed the system for the Ben Venue Laboratory where they grow and concentrate penicillin. In the concentration process they use amyl acetate (Table 1) at 400 parts per million which is a fire hazard and capable of producing dermatitis. Amyl acetate is heavy. It says here in Table 2, "dilution with uncontaminated air." It also shows a recommended method, respiratory protection. Dilution with uncontaminated air in this case had to be so handled that the air supply would drift through the working zone. The occupants would get the fresh air and the contaminated amyl acetate air would be exhausted from the bottom of the room. The exhaust air in this case was taken from the floor at one end of the room, the supply was a very low velocity drifting supply at the opposite end of the room, and that has proven very satisfactory. Temperature control was not needed and outside air is used for ventilation.

These two examples prompt me to say that any of us who are in the heating, ventilating, and air conditioning business are well advised to study this table and to make some inquiry as to which of these solvents, if any, is going to be used in these enclosed spaces.



**1258**

## INDUSTRIAL EXHAUST VENTILATION IN INDUSTRIAL HYGIENE

By ALLEN D. BRANDT,\* CHICAGO, ILL.

ONE OF the more important phases of industrial hygiene, if not the most important, is atmospheric sanitation. The air, like water, milk, foods, etc., must be relatively free of harmful contaminants if the health of the individuals breathing the air is to be maintained at a high level. Even though most air borne contaminants (dusts, fumes, mists, smokes and gases) are toxic to a greater or lesser extent, a certain amount of atmospheric contamination is permissible without affecting adversely the health of the exposed individuals.

The permissible amount of contamination varies from individual to individual and from substance to substance, but for a given substance the health of the average individual (considered as a group) is not affected adversely by breathing air containing a given amount of the contaminant.<sup>1,2</sup> For some materials the safe amounts have been determined rather accurately and for other substances a tentative value has been established on the basis of comparable chemical structure, comparable physiological reaction, or field experience. These safe limits are called maximum allowable concentrations and are usually referred to merely as M.A.C. The M.A.C. of a given substance might be defined as the atmospheric concentration of the substance which will have no measurable deleterious effect upon the average individual who breathes the contaminated air eight hours daily. This does not mean that no workers exposed to atmospheric contaminants in the range of the maximum allowable concentrations will show ill effects. Some individuals display unusual susceptibility to certain substances and will show signs of illness even if they work in atmospheres which are not contaminated in excess of the maximum allowable concentration. In general, however, if the atmospheric contamination in industry does not exceed the maximum allowable concentration for the substance or substances in question, the large majority of the exposed workers will show no demonstrable ill effects.

The maximum allowable concentrations vary tremendously from substance to substance. The units in which these concentrations are expressed vary also from substance to substance but, in general, are similar for similar types of substances or types of physiological reactions. Thus the M.A.C. for hydrogen sulfide is 20 ppm; for gasoline 1000 ppm; for lead the M.A.C. is 0.15 mg/m<sup>3</sup>, and for TNT it is 1.5 mg/m<sup>3</sup>.

Experience in a number of industries has shown that (1) occupational illnesses of a respiratory or systemic nature are rare if workers are not exposed to atmospheric contamination in excess of the maximum allowable

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<sup>1</sup> Superior numerals refer to References.

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concentration; (2) the incidence of occupational illness increases rapidly if workers are exposed to excessive amounts of atmospheric contamination; and (3) the atmospheric contamination can be kept below harmful limits by engineering measures.

Atmospheric contamination in industry is more common than most people realize. Many persons feel that so long as the air is not visibly contaminated or does not cause irritation to the nose and throat, it is respirable, that is, safe to breathe. Consequently, they feel that only relatively few operations or processes are of concern from the point of view of atmospheric contamination. This is, of course, erroneous, for the air may appear perfectly clean and may be wholly free from any irritating action even though it is contaminated greatly in excess of the maximum allowable concentration. It must be remembered that wherever materials are being processed, whether changing the physical or chemical state or the physical size, some of the material will escape into the workroom air in the form of dusts, fumes, mists or gases, except where the operations or processes are conducted in air-tight systems or in closed systems maintained under negative pressure. Therefore, consideration must be given by engineers to the problem of atmospheric contamination by most processes and at most operations rather than those few operations or processes which produce visible contamination.

The measures commonly employed for the prevention of excessive atmospheric contamination in industry are as follows:

1. Control of the dust, fume, mist, smoke, or gas at the point of generation or dissemination. *a.* Local exhaust ventilation; *b.* Enclosures; *c.* Wet methods; *d.* Good housekeeping.
2. General ventilation.
3. Isolation or segregation of the operations producing atmospheric contamination.
4. Substitution of less toxic materials for the toxic ones or of less objectionable operations for the ones producing much contamination.

Of these measures, local exhaust ventilation and general ventilation are by far the most important. Consequently, atmospheric sanitation falls largely within the purview of ventilating engineers, and most of this paper will be devoted to a discussion of some of the primary factors involved in the design of adequate, efficient, and economical industrial exhaust ventilating systems.

#### LOCAL EXHAUST VENTILATION

Local exhaust ventilation, sometimes referred to by industrial ventilating engineers as process ventilation, is usually employed to control the dust, fumes, gases, or mists at specific and major sources of contamination, whereas general ventilation is applicable to areas having scattered minor sources of contamination. General ventilation is unsatisfactory for the control of important sources of contamination, particularly if workers are nearby, since the ventilation rate must be enormous to result in adequate dilution of the contaminated air between the source of contamination and the worker's breathing zone.

A local exhaust ventilating system usually consists of four main parts—hoods or enclosures, ducts, exhausters, and collector or collectors. Of these, the hood is probably the most vital. It is the purpose of the hood to enclose or partially enclose the source of contamination or to produce air movement at the source of contamination of suitable magnitude and acting in the proper

direction to capture the escaping contaminants and convey them into the exhaust system. A thorough knowledge of the laws of air flow into suction openings and of the way in which the various contaminants react is essential to the design of an efficient hood.

While the design of a good hood involves many considerations, the following specific rules should be kept in mind at all times: (1) Enclose the source of contamination as much as possible, (2) locate the hood in line with the natural direction of movement of the contaminant or contaminated air, (3) locate hoods which do not enclose the sources of contamination, as close to the sources as possible, and (4) for hoods which must be located at some distance from the source of contamination, use large hood openings and flanges if possible.

The velocity of air movement at the source of contamination necessary to capture the contaminant may vary from as little as 75 fpm, a velocity

TABLE 1—MINIMUM AIR VELOCITIES RECOMMENDED FOR THE CAPTURE OF DUSTS, FUMES, MISTS, GASES AND VAPORS RELEASED IN CERTAIN MANUFACTURING PROCESSES

CONDITIONS OF GENERATION OF CONTAMINANT	RECOMMENDED MINIMUM VELOCITY FT./MIN.	EXAMPLES OF PROCESSES
Released without noticeable air movement.....	75-100	Evaporation or escape of liquids from open vessels such as degreasing, pickling, or plating tanks; manual handling of small amounts of dry materials.
Released with low air velocity	100-200	Spray paint booths, cabinets and rooms; dumping dry materials into hoppers; welding.
Active generation.....	200-500	Some spray painting operations in small booths and with high pressures; active barrel filling; loading conveyors.
Released with great force....	500-2000 and higher	Grinding; abrasive blasting.

sufficient to overcome normal air currents, to as high as 2000 fpm or more, at a high velocity dust producing machine such as a *jack hammer*. Table 1 will serve as a guide in the selection of the proper minimum control velocity for any operation, and shows that the minimum control velocities for the great majority of operations are in the range of 75 to 300 fpm. Even though these velocities will do a satisfactory job if the operation is carried out as intended, and if other adverse influences are prevented, it is very easy to create conditions which will render the local exhaust system ineffective. For instance, any unusual motion in the area of the source of contamination will interfere with the control air currents created by the hood and render the control ineffective. Also opening doors or windows near a local exhaust hood on a moderately windy day will impair the effectiveness of the control. To obtain a clearer picture of the importance of this particular item, namely, that the operation be carried out as intended when the hood was designed and that all adverse influences be avoided, it is only necessary to reflect upon three facts; (1) most minimum control velocities are 300 fpm or less, (2) the

average person walks at the rate of about 350 fpm, and (3) the velocity of air movement through open doors and windows on a moderately windy day is in the order of 500 to 1000 fpm. Hence little disturbance is required to upset the control velocity pattern at most operations with the result that some of the contaminant will escape into the general room air. This fact is most important since it is not uncommon to find that open doors and windows near operations provided with local exhaust ventilation render such control measures essentially useless. The fault lies in the design of the plant layout—operations of this nature should not be located near doors and windows. However, with construction completed and the plant in operation, the only solution is in the erection of air barriers and the education of the worker to open doors and windows judiciously. In small rooms or bays where all operations must of necessity be located near the doors or windows, the education of the workers in the proper performance of the operation and the wise regulation of doors and windows is essential.

To obtain the velocities cited in Table 1 for different types of operations, it is necessary to know what quantity of air must be exhausted through the hood. Here again the individual circumstances must be taken into consideration but the following four equations with minor changes will serve to cover all types of operations:

For hoods which partially or wholly enclose the source of contamination,<sup>1</sup>

$$Q = VA \quad \dots \dots \dots (1)$$

where  $Q$  = quantity of air to be exhausted in cubic feet per minute.  
 $V$  = recommended minimum control velocity in feet per minute (see Table 1)  
 $A$  = total area of all openings into the enclosure through which the air enters.

Examples are laboratory hoods, spray booths, abrasive blasting cabinets, chemical reactors, and enclosures around pots or kettles containing toxic materials which are escaping into the air, such as molten lead.

It is apparent from the foregoing equation that the area of the openings into the hood should be as small as is consistent with satisfactory performance of the operation so that the volume of air required is kept at a minimum. It has been our experience that existing installations (particularly spray paint booths) which do not afford adequate control because of low control velocities may often be made adequate by decreasing the area of the opening without interfering significantly with the performance of the operation.

For canopy hoods located above tanks, tables and the like,<sup>2</sup>

$$Q = 1.4 VPD \quad \dots \dots \dots (2)$$

where  $Q$  and  $V$  are the same as in Equation 1.  
 $P$  = hood face perimeter in feet.

$D$  = distance in feet from the hood face to the table or tank top.

Examples are the hoods over steam tables, cooling kettles, and immersion tanks.

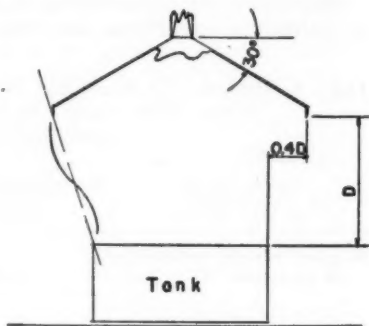


FIG. 1. CANOPY HOOD

To obtain good results, the edges of the hood should extend beyond the edges of the tank or table as shown in Fig. 1. It is essential that operations requiring canopy type hoods be located away from doors and windows since extraneous air currents influence the control air currents very readily owing to the relatively large open area between the top of the table or tank and the bottom of the hood.

Even though there are many sources of atmospheric contamination which may be controlled adequately by means of canopy type hoods, they are not so common today as they were years ago. An important disadvantage of, and objection to, canopy hoods at many operations is that the contaminated air is moved through the breathing zone of the worker and consequently does not protect the worker at the operation even though the contaminant is prevented from escaping into the general workroom air. Canopy hoods are being replaced to a great extent by slot type lateral exhaust hoods, and by down draft ventilation.

For slot type exhaust hoods such as are used at the upper edges of tanks, vats, tubs, tables, etc.<sup>4, 5, 6</sup>,

$$Q = 2.8 VLW \dots \dots \dots (3)$$

where  $Q$  and  $V$  are the same as in Equation 1.

$L$  and  $W$  = tank length and width respectively in feet.

The value of 2.8 does not apply to all conditions but may vary from about 2.0 to 3.5, as will be discussed later. Furthermore, determining a satisfactory control velocity for operations such as degreasing or electro-plating in tanks having a hood along only one long side of the tank is difficult, since to obtain the minimum control velocity at the far edge of the tank requires an unnecessarily large ventilation rate and frequently an excessive rate of solvent loss. There is much evidence available to indicate that for degreasers  $Q = 50LW$  and for electro-plating tanks  $Q = 120LW$  provides adequate control of the operations<sup>7, 8, 9</sup>. The respective minimum control velocities are about 18 and 43 fpm, which are too low to be of much significance. The reason for the satisfactory results is, of course, that the velocity is much higher over most of the tank and that the mist or vapor that does escape from the far side of the tank is not sufficient to contaminate the nearby air excessively.

Whenever slot type hoods are used on tanks, table tops, and the like, they should be located at least on both long sides of the tank and preferably along the entire perimeter. Even with the same total ventilation rate, better control is accomplished since a higher velocity is maintained at the tank edge, where interference from extraneous air currents created by movement of nearby workers is the greatest.

For unrestricted hoods at a short distance from the contamination<sup>3</sup>,

$$Q = V(10x^2 + a) \dots \dots \dots (4)$$

where  $Q$  and  $V$  are the same as in 1.

$x$  = distance in feet from the face of the hood to the source of contamination.

$a$  = area of the hood opening in square feet.

Examples are the hoods frequently used at welding, granite cutting, grinding, and similar operations. This equation is accurate only for free flow into an open hood. If the hood lies on a large flat surface the air flow is cut off from one side of the hood and the value of  $Q$  may be decreased as

much as one-third for very favorable conditions. Likewise if substantial flanges are used around the hood opening, the velocity contours will be elongated considerably directly in front of the hood and the value of  $Q$  may be decreased from 10 to 20 per cent. On the other hand, if there is some obstruction between the source of contamination and the face of the hood, the value of  $Q$  must be increased accordingly.

Tables or work benches with down draft grille tops constitute a different type of hood. An analysis of the problem will reveal, however, that such hoods fall into group 1 or group 4, depending upon whether the contaminant is released directly at the grille or at some distance above it. If released directly at the grille the value of  $x$  in Equation 4 becomes zero and the resulting equation is  $Q = Va$ , which is identical with that in Equation 1.

To paint a better picture of the fundamental laws governing the characteristics of air flow into suction openings it may be well to see how these equations fit the basic physical laws of the flow of fluids.

The first equation is obviously the fundamental law for the flow of fluids and needs no further breakdown.

To better understand the second equation, reference should be made to

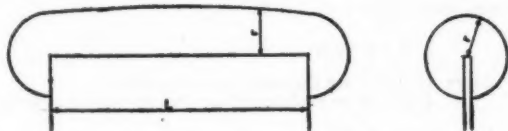


FIG. 2. SLOT TYPE HOOD

Fig. 1. All the air removed by the hood must pass through the area between the lower edges of the hood and the upper edges of the tank, that is,  $A = PD$  (approximately). Consequently,  $Q = PDV$ . However, the average velocity through this opening is considerably higher than the control velocity at the tank top. A factor is needed to have the equation apply at the source of contamination and it was found by Dalla Valle that for the usual design of tanks and canopy hoods the velocity at the upper edge of the tank is only 70 per cent of the average velocity through the open area between the tank top and the lower edges of the hood, hence  $Q = 1.4PDV^2$ .

To break down the third equation it is necessary to visualize a slot type exhaust hood as a modification of a theoretical line source of suction of considerable length, as shown in Fig. 2. Under these conditions the velocity contours in a plane perpendicular to the source of suction, obviously, would be concentric circles with the center at the point of intersection of the plane with the line. The area through which air is moved at a given velocity by the line source of suction is essentially that of a cylinder, hence  $Q = AV$  becomes  $Q = 2\pi rLV$  or  $6.28 rLV$ . It has been found by experiment that the velocity contours in a plane perpendicular to a slot type exhaust hood are essentially circular in shape but are elongated slightly directly in front of the hood opening and are flattened at the sides of the hood\* (see Fig. 3). Furthermore, with slot type exhaust hoods on tanks, tables, and the like, a considerable portion of the cylindrical surface through which the air might enter the hood is cut off so that the constant is decreased considerably and has been found to be in the order of 2.8\* (a value of 2.3 was reported by Silverman<sup>8</sup>). Consequently, since the radius of the contour is the width of the tank,  $Q = 2.8WL V$ .

To obtain a better understanding of the Equation 4 type of hoods it is necessary to visualize the nature of air flow into a theoretical point source of suction as indicated in left part of Fig. 4. The equal velocity surfaces would be spheres and the equation of air flow would be  $Q = 4\pi r^2 V$  or  $Q = 12.57 r^2 V$ . However, the contours are elongated in front of a freely suspended hood and are affected by the size of the opening (see Fig. 4). As a result, the constant is decreased somewhat and the hood area must be taken into consideration. Dalla Valle found that the value of the constant is 10 and that the correct equation is

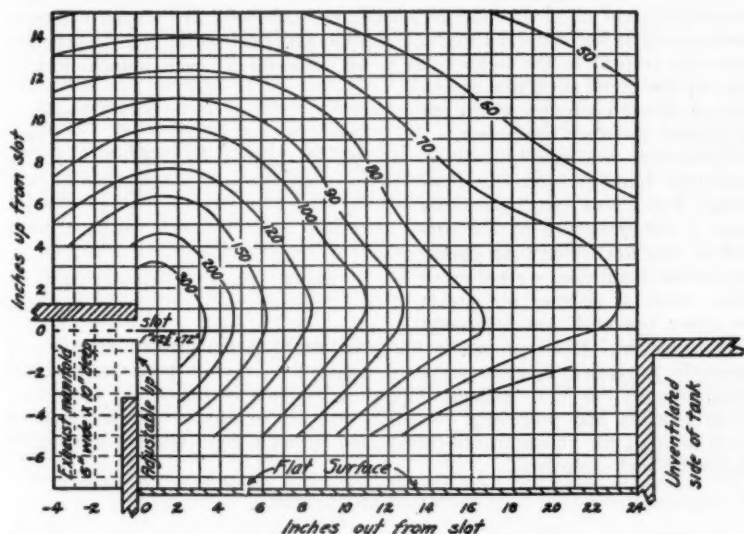


FIG. 3. VELOCITY CONTOURS FOR PLATING TANK EXHAUST SLOT

$Q = V(10x^2 + a)$  (the radius  $r$  of the approximate sphere is the same as  $x$ , the distance from the hood face to the source of contamination).

It is necessary that the foregoing fundamentals of air flow into suction openings be understood thoroughly if the engineer is to be able to design local exhaust ventilating systems which will control the contaminant effectively and economically. A much too large percentage of the installations seen by us either do not control the contaminant effectively or are so completely over designed that they are wasteful of power and present unnecessary heating problems.

The ducts should be designed almost entirely on the basis of transporting velocity which varies from about 1500 to 5000 fpm, depending upon the nature of the contaminant. As a rule, the system should be so designed that it will be balanced, eliminating the need for adjustable dampers in some branch ducts. Such dampers are a constant source of trouble. Long radius bends are desirable and branch to main duct connections should have an included angle of less than 30 deg.

Collectors are not always needed in local exhaust ventilating systems. If the contaminant is of no value, does not present a nuisance if discharged into the atmosphere, and does not present a fire, explosion, abrasion, or corrosion problem in the ducts or fan, collectors are not necessary. As a rule, collectors should be located upstream from the fan and it is sometimes advisable to have them located as close to the hood as possible for reasons of fire and explosive safety. In the explosives manufacturing and processing plants, for example, most of the local exhaust ventilating systems which have been installed in the last year have an industrial collector of the wet type located adjacent to, or as close as possible to, each hood so that the explosive contaminant is not conveyed through considerable duct work. Also particular attention is paid to the location of a wet collector in each branch duct so that in the event of a fire or explosion at one operation it will not propagate to other operations.

Exhausters are usually centrifugal type fans, or ejectors. Propeller type fans are sometimes used if the resistance of the system is very low. For most operations centrifugal fans are preferred since ejectors, whether air, steam or water operated, are inherently very inefficient. However, where the contaminant is flammable, explosive, or unusually corrosive, ejectors are used frequently. A fair percentage of the exhausters in the local exhaust ventilating systems in the Army explosives manufacturing and processing industries are air or steam operated ejectors since most of the contaminants transported through these systems are highly flammable and explosive.



FIG. 4. FREELY SUSPENDED HOOD

#### GENERAL VENTILATION

As stated previously, general ventilation is satisfactory for the prevention of excessive atmospheric contamination in rooms or buildings housing operations or processes which present minor and scattered sources of contamination. Control of excessive contamination is accomplished by introducing sufficient uncontaminated air to dilute the contaminant to a level below the maximum allowable concentration. The volume of air needed can frequently not be determined with any degree of accuracy. However, a fair estimate of the increase in ventilation needed to control excessive contamination in a given room or building may be made by determining the degree of atmospheric contamination, and estimating the existing ventilation rate. From these values the additional volume of air needed can be computed readily.

In rooms or buildings where operations are conducted involving, for example, the use of a solvent, the quantity of solvent used daily may be ascertained from the stock room records or other records which indicate the rate of consumption. With this information available, the ventilation rate in cubic feet per minute may be computed by means of the equation,  $Q = \frac{X^{10}}{M}$ , where  $X$  is the quantity of toxic substance released in cubic feet per minute and  $M$  is the maximum allowable concentration of the contaminant per cubic foot of air. For example, if a gallon of carbon tetrachloride (M.A.C. 100 ppm)

is used every 8 hour shift for miscellaneous cleaning purposes in a room of a building, the ventilation rate required to prevent the room air from becoming contaminated in excess of the maximum allowable concentration is determined as follows:

$$X = \frac{(\text{cc/gal}) (\text{sp. gr.}) (\text{mol vol})}{(\text{mol wt}) (\text{liters/cu ft})} = \frac{3785 \times 1.58 \times 24.4}{154 \times 28.3} = 33.5 \text{ cu ft}$$

$$M = 100 \text{ ppm} = 0.0001 \text{ cu ft vapor/cu ft of air}$$

$$\text{Hence } Q = \frac{33.5}{0.0001} = 335000 \text{ cu ft/8 hrs} = 698 \text{ cfm}$$

It may be well to point out that the important factor in general ventilation is the ventilation rate in terms of cubic feet per minute and not air changes per hour. This fact is not commonly recognized and engineers are frequently heard to express the thought that the recommended ventilation rates in cubic feet per minute are too high because *that will be over 20 air changes per hour*. The rate of air change means little unless the size of the room is taken into consideration. For instance, with good air distribution and all other things being equal, 10 air changes per hour in a room of a given size will result in the same degree of contaminant control as 30 air changes per hour in a room having one-third its cubical content.

#### INDUSTRIAL VENTILATION IN ARMY PLANTS

In the Army explosives manufacturing and processing plants we encountered some problems which are unique in industry and which presented new problems.<sup>11</sup> The size, location, and distribution of the buildings presented an extraneous air movement problem. The nature of the contaminant and operations presented a serious fire and explosion hazard. The nature of the industry presented an important economic problem, that is, the frequent changes in operations and anticipated short life of the industry dictated in favor of low first cost even at high operating and maintenance costs.

The problems were solved by using ventilation only where absolutely necessary and preventing excessive exposure to workers by enclosing the operations as much as possible, rotating workers, and using respirators at operations presenting intermittent exposures. Nevertheless, a large number of local exhaust ventilating systems were found necessary. The systems installed may be grouped into three general classifications, all of which have been found satisfactory from both the health and safety viewpoints. These groups are as follows: (1) Unit systems with an industrial wet collector located adjacent to, or as close as possible to, each hood. (2) Unit systems with all ductwork sloping downgrade toward a sump located outside the building, and with sufficient water sprays located throughout the ducts to keep all interior surfaces of the ducts and fan wet at all times. The fan is located downstream but upgrade from the sump so that most of the water does not pass through the fan but is drained into the sump by means of a Y in the duct at the sump. (3) Single duct systems with or without filters, and employing ejectors as exhausters.

Additional detail on these installations will not be given here but may be found in references<sup>11, 12, 13</sup>. However, a few of the results obtained will be mentioned to indicate the excellent improvement in atmospheric sani-

tation which has been accomplished. At mechanical screening or sieving operations where the workers' exposures to atmospheric dust both at the charging and discharging stations were of the order of 10 to 60 times the maximum allowable concentration without adequate control, the exposures were reduced to about one-third of the M.A.C. with local exhaust ventilating systems designed by our engineers. At a cupping operation, a reduction from 2 to 10 times the recommended maximum level to only a trace was effected. At a production box filling operation, the exposure was reduced from a level of two to four times the maximum allowable concentration to about one-third of this value.

It has been our experience that respirable air can generally be maintained economically at most operations in industry by the use of carefully planned exhaust ventilating systems. There is little excuse from the public health engineering viewpoint for the continued existence of air in industry which is contaminated sufficiently to be a health hazard to workers.

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#### DISCUSSION

L. T. AVERY, Cleveland, Ohio: I want to congratulate Major Brandt on emphasizing the control of exhaust ventilation. My pet peeve is that all discussions of ventilation that I have read in fan catalogs, manufacturers data, or the literature emphasize the exhaust requirements only, and yet, the supply requirements are of equal, if not more, importance. I believe in Equation (1) the "A" area referred to as the size of the air supply openings means the supply to the exhaust system

and not the supply air system to the room itself. If the exhaust system is permitted to draw air through an open window or an open door, then that supply is lost in the cold weather because the workers near the opening will close it. If a supply system is such that a large quantity of air enters at one point, the inertia of that entering air is of such a nature that it will set up eddy currents that will completely nullify the exhaust control. The inertia of air entering, in any direction, is a very real thing and is very much disregarded in planning of exhaust systems. An exhaust system which depends entirely on outside air is economically unsound because of the heating requirements in cold weather and the cooling requirements with air conditioning. I think the sins of the ventilation industry can be expiated as follows: provide an equivalent supply system for each exhaust system; control that supply in regard to quality and quantity of air; control its direction and recognize the inertia factor; and plan a ventilation system that will not only exhaust the dirt created in the room, but also will not bring in foreign dirt from outside with the ventilation air.

Sometimes the air coming in from the outside is as bad as the air being exhausted from the inside, because some other exhaust system contaminates the air supply.

T. H. MABLEY, Detroit, Mich.: I would just like to supplement what Mr. Avery said, and first of all to compliment Major Brandt on his suggestion that this problem is one for the heating and ventilating engineers. Many times heating engineers have trouble with the heating plant because some ventilation engineer has installed a great big exhaust system. I am recommending to the Michigan Chapter that one particular meeting, or part of our program, be set aside for the study of industrial ventilation.

As suggested by Mr. Avery a study should be made to determine what part of the exhausted air can be used to cut down the load on the heating system and to combine the ventilating and heating into one system.

W. N. WITHERIDGE, Detroit, Mich.: I am very sympathetic with all of the comments that have been made so far, and would like to add that one of the difficulties we have noticed in the Detroit area is that too many of the industrial ventilating systems are installed without engineering supervision or design. The engineers in many cases have neglected to promote this field of interest. Now there are very many large installations that go into industry for the control of air contaminants. Some of you recently have encountered them yourselves, and you know that many of these large systems are installed by the trial and error methods so popular with small installations. Most of the fundamentals, as Dr. Brandt has pointed out, remain the same, of course, for a small job as well as for a very extensive installation.

Comment was made about the lack of proper air supply to correspond with the amount exhausted. From our engineering studies in the Detroit area, it appears that almost one third of all the failures of industrial ventilation are due to the fact that adequate air supply has not been provided. We worked in this field principally with the small plants prior to the installation of large war industry ventilating systems, but recent investigations indicate that the same holds true with both the large and the small installations, so far as we can see. Some of the large jobs that are being installed today still fail to incorporate all the fundamentals of airflow that we should have learned while we were putting in the smaller systems.

The ventilating engineer has, I feel, an excellent opportunity now that there seems to be more demand for good atmospheric control in industry; he has an excellent opportunity to lead those in the field of industrial hygiene along the right paths. As you know, the engineers in the public health and industrial hygiene fields, as a rule, are not ventilation specialists. For various reasons they have too frequently struggled along with the control of atmospheric contamination without the assistance of competent ventilating engineers. The necessity for better control of the atmosphere in industry has been brought about because we know more about atmospheric contamination and its effect on health. We know that perhaps the best way to prevent occupational diseases in industry due to air contaminants is to use ventilation

properly, and we are in a period where we need expert guidance by those who are really qualified to design and install good ventilation systems.

E. C. EVANS, Buffalo, N. Y.: I have had some experience with chromic acid ventilation, and I notice in Fig. 3, showing a usual construction, that the slot is too close to the top, as it has been in all of the research that has been conducted. I would like to suggest that Major Brandt proceed with work along the same idea but apply the duct to the center of the tank, to obtain a parapet wall, also apply a submerged slot to obtain velocities multiplied by four. The effect will be a reduction in the amount of air and a reduction in the friction and the total cost of the installation.

After further research, I would like to hear more about the subject because this is one of the best papers that I have seen or read for some time. With chromic acid exposure the pretty portion of the body that is lost is the bridge of the nose. I know of one case of two young ladies working in a plant who have no bridges in their noses. That ought to lead us to understand the value of this paper. All of you who have work of this nature should take one of these papers home and study it because the air supplied must equal that exhausted from the building.

L. G. MILLER, East Lansing, Mich.: I wanted to comment on Table 1, of Major Brandt's proposal dealing with capture velocities, especially the capture velocities required where a contaminant was released with no, or very low air velocities. In any high temperature work, even at very low air velocities, the molecules of the gas, just formed from a liquid, display very much activity on their own part if they are at a high temperature. If they are heavy molecules they will have an increasing amount of kinetic energy which of course has to be overcome by the air stream. So I should like to see added to Table 1, which is the first table of capture velocities I have seen in print, some recognition of the temperature of those vats or of that contaminant, because I believe that molecular activity will play a very real part in its ability to be captured.

**AUTHOR'S CLOSURE:** I heartily endorse all the comments that have been made regarding the air supply that is necessary for make-up air. It was omitted from the paper not because it was considered unimportant or was not recognized, but because the paper was limited to one phase only. To cover all the phases of the problem would require a very long paper. The lack of make-up air is one of the important difficulties that we experience. An engineer sometimes designs a system for a small room, particularly a small room which is fairly tight in winter, and tries to exhaust a relatively large quantity of air but fails to remove much air or control the hazard. Even though workers are asked to open the doors, they keep them closed, and the system does not work. We recognize that, and we run into some very unusual circumstances and conditions owing to the fact that there is no air supply. In most of our work, in most of our recommendations, and in most of our writings, you will find that we do recommend that heated, or tempered air be provided, at the same rate, or somewhat in excess of the amount that is removed.

Regarding the discussion on locating the slot lower, it may be said that our study was more or less a fundamental research problem. Locating the slot lower in the tank or in a blank wall, or making any such change will improve the control just as though a flange had been provided along one edge. As suggested in the discussion in the paper on the nature of air flow into a slot type hood, the area through which air is drawn can be decreased to one-quarter of a cylinder. The air would then be moving through one quarter of the surface of a cylinder and the area through which the air is drawn would be reduced accordingly by that much, and consequently, as Mr. Evans pointed out, the control velocity is increased by approximately four times that of a freely suspended slot hood. That is correct, and a lot of work has been done on the problem. Silverman has published several papers on the influence of a flange, and as you will remember, earlier in my paper I pointed out that wherever possible, flanges should be used on all types of suction openings.



**1259**

## THE USE OF GLYCOL VAPORS FOR AIR STERILIZATION AND THE CONTROL OF AIR BORNE INFECTION †

By B. H. JENNINGS,\* EDWARD BIGG, M.D.,\*\* F. C. W. OLSON,†† EVANSTON, ILL.

**T**HE TOLL of air borne disease presents perhaps the largest tax on one's comfort and productive effort. Respiratory infections are responsible for an annual loss of more than 100,000,000 man days in American industry. Control and reduction of this waste represents a real challenge to engineers and physicians.

The concept of the spread of disease by the aerial route is not new, but it was not until recently that cumulative evidence has produced substantiation of this theory. Much of the credit in this field must go to the fundamental work of Wells,<sup>1</sup> who showed that droplets delivered from the respiratory tract by coughing, sneezing or talking, evaporate rapidly and become *droplet nuclei* which float in the air for long periods, finally settling to the floor in the room. Here they adhere to dust particles and can again be set into motion by air currents. The development of simple, yet reliable devices for quantitative collection of bacteria from the atmosphere has also simplified and furthered bacteriologic studies on this problem.

A number of workers have reported evidence, both clinical and bacteriologic, on the spread by air transmission of specific infections<sup>2</sup> such as hemolytic streptococcal infections, measles and mumps. Laboratory experiments have also proved that infection of susceptible animals occurs when they are exposed to atmospheres containing pathogenic organisms.<sup>3</sup>

Air-borne disease may be controlled by three methods: (1) Production of immunity in exposed individuals by vaccines, serums, drugs, etc., (2) prevention of introduction and dispersal of disease-bearing organisms into the atmosphere, (3) reduction in the number of organisms in the air of enclosed spaces.

A discussion of the first two points is not within the scope of this presentation, but it may be said that to date these measures leave much to be desired, although great strides in their development have been and are being made.

It is obvious that frequent air exchanges in a space will, by dilution, reduce the total bacterial contamination of the space. This is the commonest means of bacterial control in use. However, in addition to the fact that the method is not positive there are limits to the number of air changes possible. Filtering

† The work described in this paper was done under contract recommended by the Committee on Medical Research between the Office of Scientific Research and Development, and Northwestern University.

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<sup>1</sup> Superior numerals refer to Bibliography.

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and electric precipitation of dust and organisms in air also reduce contamination but fail to achieve complete control.

The bactericidal activity of ultra-violet irradiation is well established. If micro-organisms are exposed to the direct action of the rays, death is quickly produced. There are, however, certain objections to the widespread use of this means of air sterilization. Since exposure of the skin and eyes to ultra-violet rays may produce undesirable effects, individuals in the treated space must be shielded from the direct rays. This necessitates irradiation of only those zones in which the rays can be directed without interference with the persons in the room. Although the bacterial content of such zones is lessened, there is a remaining area in which direct droplet infection may occur and in which the bacterial counts are high. The total effect is nevertheless a reduction in air contamination since air currents ultimately expose all the air to the irradiated area.

The use of ultra-violet lamps is further restricted by cost of installation and operation. The elements must be changed at intervals because after a certain period the wavelengths of the emitted rays are no longer bactericidal. Greatest germicidal activity is exhibited when dried organisms are exposed to the rays; the effect on moist droplets is minimal. There have been, however, several reports on clinical trials of this method which suggest that under certain conditions it may be effective.<sup>4</sup>

Sterilization of air by germicidal mists was attempted many years ago by Lister, who sprayed operating rooms with phenol solutions, but further development of this method of bacterial control awaited the development of chemicals which fulfilled certain criteria. These include: (1) Non-toxicity, (2) high bactericidal activity in low concentrations, (3) imperceptibility to the sight, taste, and smell of the individual exposed, (4) availability in large quantities at low cost, (5) ease of introduction into and maintenance of concentration in the treated space.

The first practical approach to this problem was made by Douglas, Hill and Smith in 1928.<sup>5</sup> This paper dealt with the use of atomized solutions of  $\text{NaOCl}$ . They were able to produce marked reduction in bacterial counts of air-suspended *B. coli* using a concentration of 0.5 mgm  $\text{NaOCl}$  per liter of air.

Ten years following this report, Masterman<sup>6</sup> extended these earlier observations and reported sterilization of air by 0.02 mgm of 1 per cent  $\text{NaOCl}$  in 1 liter of air. He stated that the germicidal effect was produced by molecular  $\text{HOCl}$  liberated from the sprayed droplets of  $\text{NaOCl}$ . A concurrent report by Trillat<sup>7</sup> appeared in 1938. After experimentation with the atomization of the common germicidal agents he concluded that only resorcinol and sodium hydrochlorite were satisfactory. Trillat also propounded the theory, which is now questioned, that these substances exert their effect because of their physical state, i.e., as aerosols. In brief, he believed that each droplet of sprayed bactericidal compound retained the concentration of the parent solution and because of its small size (1 to 2 microns) remained suspended in the atmosphere, where it ultimately collided with droplets containing bacteria, thus acting in the same concentration as the original solution.

Continuing observations were carried farther by certain English workers. Pulvertaft and Walker<sup>8</sup> advised solutions of resorcinol, glycerol, and water. Twort<sup>9</sup> and his associates after a comprehensive study of the subject developed a solution containing hexyl resorcinol, loral sulfate and alkaline propylene glycol which they found highly effective. Both groups of investigators

based their experiments on Trillat's aerosol theory and for this reason attempted to find means for the prevention of evaporation of droplets. This explains the first use of glycerol and propylene glycol which because of their low vapor tensions and hygroscopicity maintained droplets in air for longer periods.

#### PROPERTIES OF THE GLYCOLS

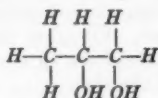
Recently the glycol group of compounds in vapor form have been found to be highly effective agents for air sterilization.<sup>10</sup> Although all glycols possess killing power in varying degrees, propylene glycol (PG) and triethylene glycol (TEG) fulfill most completely the criteria noted above for widespread use. These materials had been used industrially as solvents and dehumidifying agents for many years, but it was not known that they presented any germicidal value. In fact, they exhibit relatively low bactericidal activity in test tube experiments. Cultures of micro-organisms grow readily in broth containing 15 per cent propylene glycol and death does not occur until the glycol concentration of the media reaches 75 per cent or over. However, when dispersed in the air, minute quantities in vapor form exhibit a dramatic killing action. It might be mentioned at this time that this latter observation tends to disprove Trillat's aerosol theory. Since death is produced almost instantaneously in air suspended organisms, this phenomenon cannot be explained by collision of bacterial and glycol droplets, but may be computed on the basis of interaction between molecular glycol and bacterial droplets.

#### Toxicity Studies

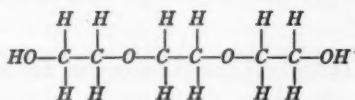
PG and TEG have the lowest toxicities of any compounds suggested for air sterilization. Comprehensive studies on ingestion and parenteral administration show that the toxicity levels are somewhat below that of ethyl alcohol.<sup>11</sup> Observations on exposures of rats, guinea pigs, and monkeys to prolonged periods of inhalation (approximately two years) have shown no local or systemic evidences of toxicity.

#### Physical Characteristics of PG and TEG

Propylene glycol,  $C_3H_6(OH)_2$  has a structural formula



with a molecular weight of 76.06, boils at 368 F and has a vapor pressure of 0.016 mm of mercury at 68 F. Triethylene glycol,  $C_6H_{12}O_2(OH)_2$  has a structural formula



with a molecular weight of 150.17, boils at 548 F and has a vapor pressure of 0.00083 mm of mercury at 68 F. It can thus be seen that these are high-boiling-temperature dihydroxy-alcohols. They are freely miscible with

water and highly hygroscopic. They are essentially odorless, both in liquid and vapor form. In very high concentrations the vapors give an impression of sweetness. Because of the non-toxic character of *PG*, which was first established, much of the early experimental work involved the use of this material. However, when *TEG* was also shown to be non-toxic,<sup>11</sup> it almost completely displaced *PG* for bactericidal use as only about one fiftieth as much *TEG* is required for the same bactericidal effect.

### Fire Hazard

The inflammability characteristics of *PG* and *TEG* have been studied and reported elsewhere.<sup>12</sup> It has been found that:

1. The vapor-phase concentrations required for air sterilization were completely free of any fire hazard.
2. The presence of water in combination with glycol greatly reduced its combustibility. This was particularly significant, since samples of condensed material from surfaces as cold as -20 F never contained more than 20 per cent glycol. It is impossible to ignite such solutions even with prolonged application of heat.
3. The introduction of small quantities of water with glycol renders large storage tanks relatively inactive as far as inflammability is concerned.

### SMALL CHAMBER EXPERIMENTS<sup>10</sup>

These experiments were carried out in small glass chambers of two cu ft capacity. Bacterial cultures were sprayed into the space by glass atomizers and measured quantities of air were removed for quantitative determination of bacterial content. Since there is a constant spontaneous reduction in recoverable organisms from the atmosphere after spraying, a control chamber was used in each experiment. Agitation of the air to insure mixing was accomplished by the use of small revolving fans. Organisms tested and found to be susceptible to the action of glycol vapors were pneumococci, hemolytic and non-hemolytic streptococci, staphylococci, *Escherichia coli*, *H. influenzae*, *B. pertussis*, and *Streptococcus viridans*. Careful controls were carried out to show that actual death for the test organisms occurred.

To demonstrate the virucidal activity of *PG* and *TEG*<sup>18</sup> mice were placed in the test chambers and a mouse-adapted strain of influenza virus (*P.R.8*) was then atomized into the space. Complete protection of the exposed mice occurred in the glycol treated chambers in contrast to almost 100 per cent death of mice in the untreated chamber.

Conclusions drawn from these experiments were: (1) In concentration of 0.2 mgm *PG* per liter of air and 0.005 mgm *TEG* per liter (0.00218 grains per cu ft)† of air immediate death of air-borne micro-organisms is effected, (2) relative humidities of 25 to 60 per cent are necessary for optimum activity, and (3) moist droplets are more susceptible to glycol actions than dried organisms.

### TESTS IN LARGE UNOCCUPIED SPACE

Studies preliminary to large scale application were conducted in an air conditioned test room of the Technological Institute of Northwestern Uni-

† For further clarification this may be expressed as an average bactericidal concentration of 0.67 parts per million by volume or 3.42 parts per million by weight.

versity shown in Fig. 1. This room is 38 ft long, 17 ft wide and 16 ft high, with an approximate volume of 10,000 cu ft. It is insulated, extremely tight in construction and with special sealing reduces air infiltration and exfiltration to a low minimum. Large refrigeration type doors were used for entry and over each of the windows. This enabled us to carry out our experiments without the factor of large quantities of uncontrolled air. Tests were made to: (1) Demonstrate the lethal action of glycol vapors in this large space, (2) test the effectiveness of vapors produced by different

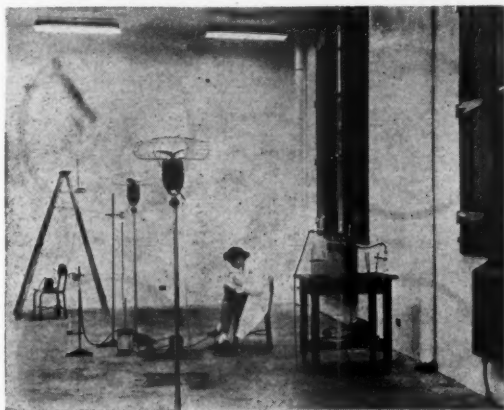


FIG. 1. EXPERIMENTAL TEST ROOM

methods, and (3) check the distribution of glycol vapors in a room of significant capacity.

The results of these experiments have been previously reported,<sup>13</sup> but it would be of interest at this time to review the findings.

Test organisms used were *Staphylococcus albus* and a guinea pig strain of hemolytic streptococcus (streptococcus C). Bacterial suspensions were made in filtered saliva. Saliva was used as the diluent since it was noted that organisms suspended in this material remain in the air for long periods of time. This simulated field conditions as closely as possible. The suspensions were sprayed by means of an atomizer which produced an extremely fine mist.

Bacterial samples were obtained by a Moulton air bacterial sampler<sup>14</sup> and reported as numbers of colonies per cubic foot of air.

A test was run as follows: The room was first thoroughly aired by opening the windows and doors and turning on two large propeller-type circulating fans placed in the room. It was then made air tight and 10 cu cm of the bacterial suspension were sprayed directly under the two circulating fans, at opposite sides of the room. Bacterial air samples were taken immediately after the cessation of the spray and at intervals of 15 min for a total period of 2 hours. The doors and windows were then reopened, fans speeded up and the room allowed to air for one hour. The space was again sealed,

glycol vapor introduced, and 10 cu cm of the same bacterial suspension used for the control test were sprayed under the fans. Samples of air with the same time intervals as before were taken to determine bacterial content. Glycol concentrations existing in the air were also measured. Results of the tests corroborated the small chamber experiments and the same striking bactericidal activity of PG and TEG vapors was demonstrated. The previous

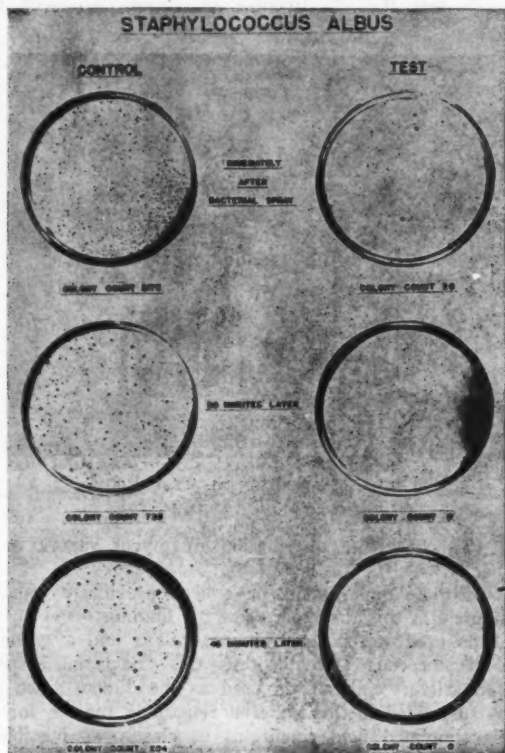


FIG. 2. BACTERIAL PLATES FROM TEST ROOM (STAPHYLOCOCCUS ALBUS)

observations on relative humidity requirements were also substantiated. At levels of 80 per cent or over there was practically no lethal effect. Fig. 2 is a photographic reproduction of a typical test.

Carefully conducted observations on the behavior of the vapors were made and showed that if air exchanges were kept at a minimum, the glycol concentration decreased at a slow rate. Data on propylene glycol show that it took more than 4 hours in the test room for the glycol concentration to reduce from 0.22 mg per liter of air to half that amount. There was

evidence of a slight amount of absorption on room surfaces in addition to the larger amount removed under the small but existent air exchange. Only when excessive concentrations of the vapor were produced was the presence of the material perceptible to the laboratory workers. Observations were made to determine whether any evidence of stratification of glycol vapor at different levels in the quiescent room existed. Tests running over a 30-hour period showed no significant variation in glycol concentration at different levels in the test room.

#### MEANS FOR GENERATION OF VAPOR

Before application of the germicidal effect of these vapors could be made, it was necessary to develop practical means for their generation. Such

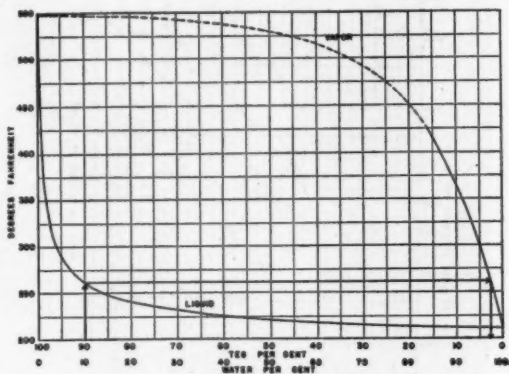


FIG. 3. TEMPERATURE COMPOSITION DIAGRAM OF TRIETHYLENE GLYCOL

apparatus had to be of sufficient size and capacity to treat large spaces and incorporate control devices so that adequate sterilizing concentrations could be built up and maintained.

The problem of putting glycol vapor into the air can be solved in several different ways, one of the most satisfactory of which was found to be by vaporization from an aqueous solution. TEG is extremely hygroscopic and miscible with water in all proportions. As is true of miscible binary mixtures, the temperature at which boiling takes place varies with the concentration of the particular mixture. At atmospheric pressure, solutions high in water will vaporize at a temperature above, but close to, 212 F. Solutions of TEG rich in glycol and low in water will vaporize closer to the boiling temperature of pure glycol, 548 F. When a glycol-water solution is heated, water, being the more volatile component, vaporizes more readily; however, both water vapor and glycol vapor are delivered from the boiling mixture. Reference to the composition diagram of Fig. 3 shows, for example, that an aqueous solution of TEG with a boiling point of 262 F delivers vapor containing about 3 per cent glycol and 97 per cent

water. This water-vapor delivery does not constitute a disadvantage but is desired since, as has been noted above, relative humidities ranging between 25 and 60 per cent are required for optimum bacterial action in the space which is to be treated with glycol vapor.

As boiling of such a mixture continues and water leaves at a more rapid rate, the remaining liquid becomes richer in glycol and its boiling temperature rises. If the heat supply is continued ultimately pure glycol will remain. From a given vaporizer vessel, in an operating range of 240-290 F, the amount of glycol removed is so small that it requires infrequent replacement while water must be added almost continuously.

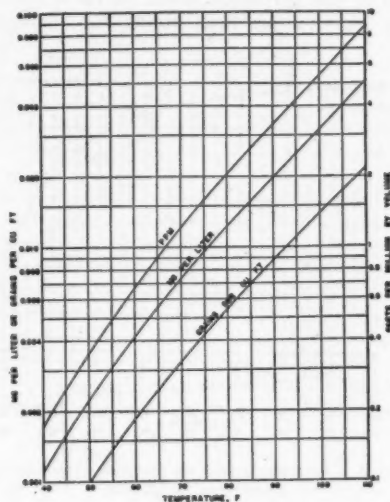


FIG. 4. SATURATION DENSITY OF TRIETHYLENE GLYCOL VAPORS

The concept of maintaining a constant glycol concentration became the basis of the control system for the vaporizer.\* As the device operates, and its temperature rises to a given point, a thermostat set at that temperature opens a solenoid valve which in turn feeds water into the vaporizer in amount sufficient to lower the vaporizer temperature to the point at which the valve is closed. Thus by action of a simple thermostat controlling a water feed valve, the boiling temperature can be kept in a narrow range and the relative rate of water and glycol delivered to the air accurately controlled. Varying the rate of heat input controls the rate at which water and glycol are delivered to the air, but does not vary the ratio of glycol to water vapor. Any desired ratio can easily be obtained by merely adjusting the temperature at which the thermostat is set to operate. A high thermostat setting gives a larger percentage of glycol vapor, a lower thermostat setting, a relatively larger

\* Patent 2,369,900 assigned to the Secretary of War.

percentage of water vapor. The functions are thus performed in this manner: *change thermostat setting to vary relative water-glycol vapor delivery, change heat input to vary quantity delivered.*

The simplicity of this control and its effectiveness have been well demonstrated in field tests in which the authors have been able to build up or reduce the glycol concentration in the space by application of the principles outlined above. With constant wattage input at the electric heater it should be observed that an increase in thermostat setting will permit the heat to

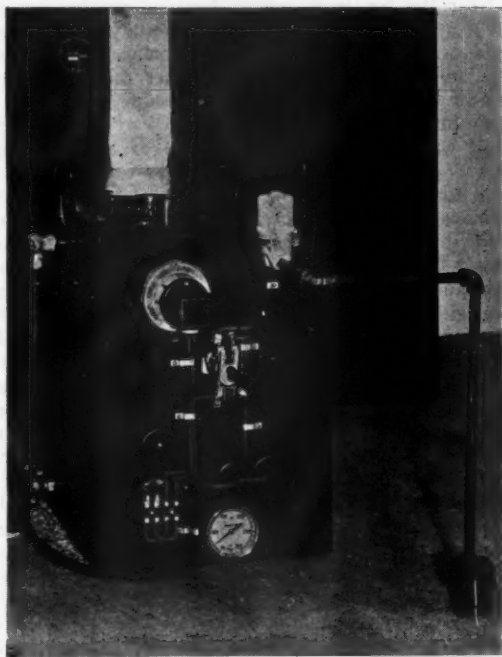


FIG. 5. GLYCOL VAPORIZER

work to a greater extent on glycol vaporization, and consequently build up the glycol concentration in the space, although under these conditions less water is supplied.

In all authors' field tests a vaporizer was used. The first tests were made using a laboratory model vaporizer, but the later tests were made with a commercially manufactured vaporizer made to authors' specifications. This is pictured in Fig. 5 and consists of a double shelled unit with the space between the shells filled with suitable insulation. The vapor is discharged from a pipe which attaches into the vapor space of the unit just beneath a removable top. This outlet can be piped to lead the vapor into a duct system or if the vapor is delivered in front of a fan it can be distributed by air

currents throughout a small room. An indicating thermometer is placed at the lower front of the unit and above this is a 1000 watt immersion heater. To the right of the heater is a temperature control switch which is adjusted to open the solenoid valve. Thus when the temperature rises to a certain point, indicating that the water concentration in the solution is too low, this switch opens the solenoid valve, which remains open until the boiling temperature drops to a low enough point for the temperature control switch to operate and close the solenoid valve. At the left of the heater element is a high-temperature safety cut-off switch to protect the vaporizer from overheating. Near the top in a central location can be seen a high-level liquid float control which trips the electric circuit in the event that the liquid level becomes too high. Below this is a manual reset which has to be used to put the unit back in service when either the float switch or high-temperature cut-off operates. A gage glass to show liquid level is attached.

In addition to investigating the characteristics of a vaporizer,<sup>15</sup> a large amount of work was done concerning the use of an atomizing device or scrubber unit for putting glycol into the air.<sup>16</sup> Under this method an atomizing nozzle forces a mist of the aqueous glycol solution into a moving air stream. Eliminator plates following the nozzle remove the liquid particles, which drain back to the pump reservoir. Water and glycol vaporize into the air stream and are delivered by the attached duct system to the treated space.

There is a certain point at which glycol vapor will be sufficient in amount to saturate completely the atmosphere. Above this point if additional quantities of glycol are present they must exist in suspension in a foglike condition. In the case of the atomizing device the amount of vapor which passes into the air will depend upon the air temperature, and a fog in the treated space would not arise unless the air temperature leaving the atomizer drops appreciably. For any aqueous solution being sprayed into passing air there is a certain relative humidity of water and glycol vapor concentration in equilibrium. The control of such a unit thus hinges about the use of the proper aqueous solution to give the desired glycol-humidity equilibrium in the treated space. Automatic controls for this purpose are being worked on but have not yet been perfected. It will be realized that air with a low glycol vapor content entering can carry only a certain maximum content of glycol vapor as it is discharged. Thus with such a system, using recirculated air and an atomizer, the glycol concentration in the air will rise to the desired maximum which can be reached only by vaporization into the airstream for the particular air temperature at which operation is being carried on.

The development of a sensitive means of detection of glycol in the air which would control the operation of the glycol generator is necessary for complete maintenance of desired room concentrations. Such devices are being studied. Until they are available one must compute the approximate number of air exchanges and introduce vapors at a rate sufficient to compensate for exfiltration. It is believed, however, that the final development of the air scrubber will operate to maintain constant concentrations.

In Fig. 4 are data on air saturated with triethylene glycol. Saturated in this sense means that the amount of vapor held is the maximum which can exist in space as vapor at a given temperature. If such saturated air is cooled, fog or mist would appear as the excess vapor over that required for saturation at the lower temperature condenses out as fog or precipitates on surfaces. Notice that the saturation density rapidly increases with temperature. Fortu-

nately, under desired inside temperatures, the bactericidal concentrations of the glycols required are far below the saturation or fog conditions in air and the glycol cannot normally be detected by the occupants in any way.

Propylene glycol, as before mentioned, is also an effective air sterilizing agent, but it is needed in much greater quantities, namely, between 0.1 to 0.3 milligram per liter (0.044 to 0.13 grain per cu ft) for effective bactericidal action. These concentrations also are far from saturation so that it can be used with little probability of detection by the occupants of a treated space. However, inasmuch as from 40 to 60 times more propylene glycol is required than triethylene, the latter is replacing it in favor.

A few simple mathematical considerations will be helpful in understanding the manner in which glycol concentrations may be built up in a room.

For example, when:

- $w$  = pounds weight of glycol vapor in room,
- $a$  = number of air exchanges per hour,
- $b$  = glycol output of vaporizer in pounds per hour,
- $t$  = time in hours,

then

$$\frac{dw}{dt} = b - aw,$$

the solution of which, assuming an initial condition where  $w = 0$  when  $t = 0$ , is

$$w = \frac{b}{a} (1 - e^{-at}).$$

The maximum value occurs, therefore, when

$$w = \frac{b}{a}.$$

As long as there is some air exchange taking place in the room, the glycol concentration will increase to a fixed value but cannot increase indefinitely. Also the greater the number of air exchanges, the sooner the final concentration will be reached.

#### TESTS ON LARGE OCCUPIED SPACES

For this purpose eight dormitories were selected, each housing approximately 80 men. Four of these were used as test spaces and four were used as controls. Each room was approximately 120 ft long, 30 ft wide, and 9 ft high, having a capacity of approximately 33,000 cu ft. A double row of lockers extended the length of the room, effectively dividing it in two. It was, therefore, necessary to use a duct system to distribute glycol and water vapor throughout the space.

Since the distribution of glycol vapors in a duct system permits some new engineering problems, a full scale model was constructed and tested in the Institute laboratory before field installations were made.

#### Laboratory Tests

The duct used in these tests was of laminated asbestos construction, specially treated on the inside to resist absorption of glycol. It consisted of three 33-ft sections  $8\frac{1}{2}$  in. x  $14\frac{1}{2}$  in.,  $8\frac{1}{2}$  in. x  $8\frac{1}{2}$  in.,  $8\frac{1}{2}$  in. x  $5\frac{1}{2}$  in., inside dimensions. The transitions from one size to another were of truncated

pyramid shape, about 6 in. long. The far end of the duct was closed and the near end connected to the fan unit as shown in Fig. 6. Three turning vanes were placed in the right angled elbow to straighten the air flow into the duct.

Instead of the customary louver and baffle arrangements, special *venturi* openings were used for distributing the air-water-vapor-glycol mixture. These openings were plaster cast fittings having a  $2\frac{1}{2}$  in. diameter free opening, faired outward and inward to give a venturi-like cross section. The duct

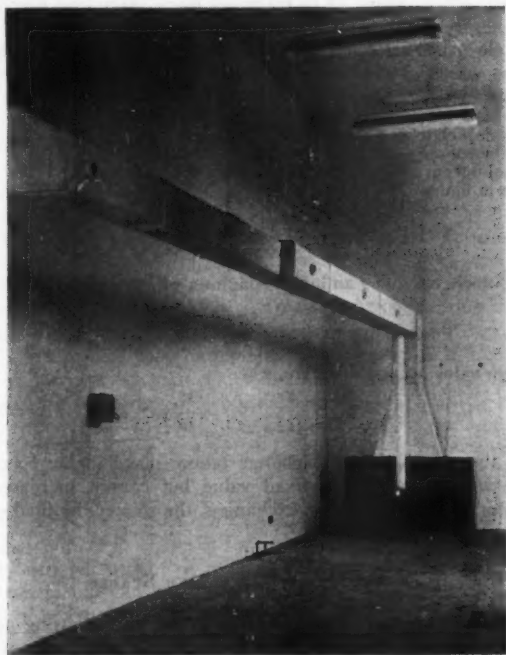


FIG. 6. ASSEMBLED DUCT FOR GLYCOL DISTRIBUTION  
(TEMPORARILY ASSEMBLED FOR PHOTOGRAPHIC PURPOSES)

was equipped with 36 such outlets, 18 on each side, spaced at approximately 5-ft intervals. Starting from the elbow, the first outlet was placed 10 ft 4 in. from the inside turn of the elbow on the right side of the duct; the second outlet was placed one foot farther on the left side of the duct.

The arrangement of the fan unit may be observed in Fig. 6. It consisted of three co-axially mounted blower fans directly connected to a  $\frac{1}{4}$  hp motor. The six speeds of the motor produced from 750 to 1700 cfm at free delivery of the fan. The speeds were controlled by a selector switch on the side of the cabinet. The front of the unit is provided with a louver and damper for controlling recirculated air. The back is connected to the fresh air intake and likewise provided with dampers. Both sets of dampers are operated

simultaneously by a shaft extending through the top left side of the cabinet. With these controls the fan output can be varied uniformly from 100 per cent outside air to 100 per cent recirculated air. A heating element was also provided in case it became necessary to heat some or all of the incoming air. In these tests the heating element was not used. Provision was also made for inserting air filters just above the heater.

The first test was on fan output with adapter and elbow. This corresponds roughly to finding free delivery characteristics at the entrance to the duct. A short length of duct (8 in.) was attached to the elbow and anemometer readings taken at various fan speeds. The results are given in Table 1.

To obtain the relative air velocities through the various outlets, anemometer readings were taken with the back of the anemometer case flush with the

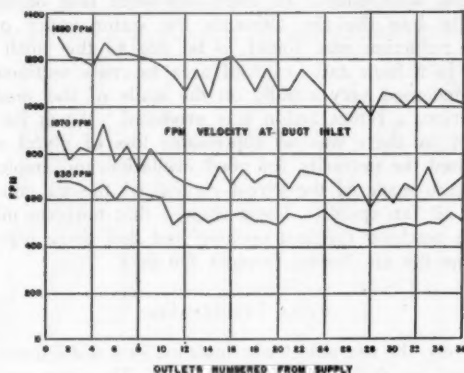


FIG. 7. ANEMOMETER DATA ON OUTLETS

opening. A true air speed reading is impossible under such conditions since the outlet is smaller than the case, but an accurate comparison of the relative outputs is readily made by this method. The results for various fan speeds are given in Fig. 7. The numbers in the chart refer to the outlets in progressive order from the fan end. It can be seen that fairly uniform air velocities are obtained. The system was further checked by the use of a smoke bomb. This showed the air to be blown 20 ft horizontally from the outlets before an excessive diffusion occurred. Since the openings were

TABLE 1—FAN OUTPUT UNDER FREE DISCHARGE

SPEED NO.	AVERAGE AIR VELOCITY (FPM)	OUTPUT (CFM)
1	1998	1690
2	1369	1160
3	1118	950
4	1000	850
5	865	735
6	830	706

only 5 ft apart, it seems certain a uniform air distribution in the room was produced.

Velometer and draft gage readings were also taken at each outlet and for each fan speed. The results are in substantial agreement with the anemometer readings. The air velocities in the duct immediately after the right angle turn computed from velometer readings at the outlets were found to be 1680 fpm for speed No. 1, 1070 fpm for speed No. 3, and 840 fpm for speed No. 5.

The glycol-water vapor output of the laboratory model vaporizers was determined by weighing the vaporizers with their contents before and after a run of 48 to 72 hours. The vapors escaped into the open air through a  $\frac{3}{8}$  in. pipe size tapping in the cover. However, when the same vaporizers were connected to the duct by a 4-ft length of well-insulated  $\frac{3}{8}$ -in. pipe, the glycol output was reduced to about one-tenth that obtained when discharging directly into the air, although the water vapor output was not affected. This reduction was found to be due to the small pipe diameter which resulted in a high ratio of wall area to cross sectional area. Since glycol vapor condensed very rapidly on the walls of the small pipe, despite adequate insulation, a reflux action was produced. When the pipe size was increased to  $1\frac{1}{4}$  in. there was no appreciable loss of glycol output. All of the work indicated the necessity for good insulation and ample pipe size.

Analyses<sup>17</sup> were made of the glycol content of the air coming from each opening and at all fan speeds. These showed that uniform mixing of glycol occurred before reaching the first opening and that there was no perceptible loss of glycol as the air flowed through the duct.

#### FIELD INSTALLATION

For the field test, the test model was installed in a test dormitory and similar units constructed for three other dormitories. The only modification made was the replacement of the first 33-ft length of duct with one having an  $8\frac{1}{2}$  in. x  $11\frac{1}{2}$  in. cross-section instead of  $8\frac{1}{2}$  in. x  $14\frac{1}{2}$  in., as in the test model. The appearance of the installation is represented diagrammatically in Fig. 8. For clarity, the double row of lockers in the center of the room and several bunks in the left foreground are not shown. The duct was hung on the center beam and fresh air brought in through an adapter placed in the window frame. A water line was connected to the solenoid valve in the vaporizer.

For adequate comfort it was decided to allow not less than 25 cfm outside air per individual. Since it was estimated that the uncontrolled normal infiltration of air was equivalent to two air changes per hour, the fan speed was set to deliver between 1000 and 1200 cfm of outside air. As far as possible, the doors and windows of the test rooms were kept closed and all air entering the duct was *glycolized*. Ventilation in the control rooms was at the discretion of the occupants; it is believed that the number of air exchanges were greater than in the test quarters.

Since the volume of the room was 33,000 cu ft with the equivalent of four air changes per hour, it was calculated that a glycol delivery of 0.04 lb per hour would be needed to maintain a concentration of approximately 0.004 mgm TEG per liter of air. To obtain this output the operating vaporizer tempera-

ture was set at 280 F. This setting produced a concomitant water delivery of 2.5 lb of water per hour.

The dormitories were heated by steam radiators with a single outside-inside thermostat controlling the flow to four rooms. The buildings used were so arranged that each thermostat regulated the temperature of two test and two control spaces. This presented a difficult problem in the conduct of the studies, particularly so as there were no individual cut off valves on the room radiators. Thus the maintenance of adequate glycol concentrations was frequently interfered with by unauthorized opening of windows because of excess temperatures.

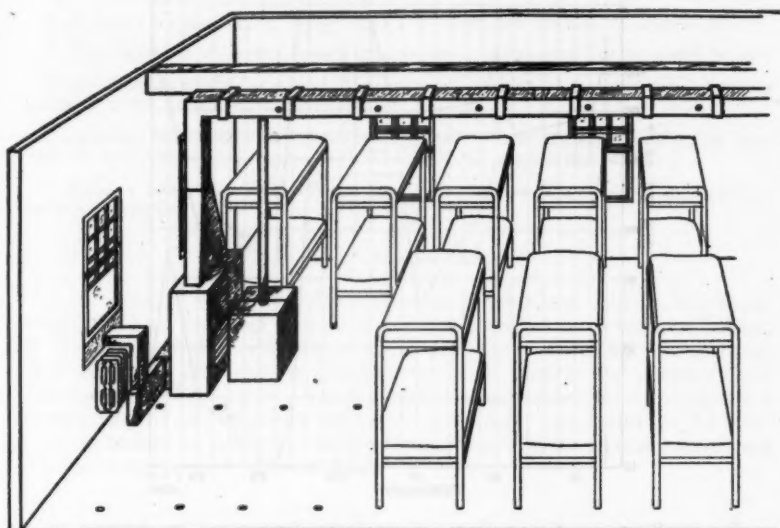


FIG. 8. FIELD INSTALLATION FOR AIR STERILIZATION IN DORMITORY

It might be noted at this time that the room temperature in the treated space averaged 2.9 F higher than in the control space and ran between 74 F and 76 F during the greater part of the time.

### Results

Some operating difficulties were encountered, but these were not serious. It was necessary to add glycol to the vaporizers at three- to four-week intervals and no other servicing was required. Condensation and resultant dripping of liquid glycol from the first duct section occurred at irregular intervals and was most disturbing during periods of extremely cold weather. Tempering of the incoming fresh air would have eliminated this trouble. However, due to the unsatisfactory control of heating system in use this could not be done.

Anemometer readings on air flow and glycol output from various duct openings agreed with the laboratory observations. Glycol distribution in the

rooms was uniform. Glycol determinations were made at 2 a.m., 5 a.m. and 8 a.m. The average concentrations were found to be 0.002 to 0.003 mg per liter of air. These values would probably have been higher if duct condensation had not occurred and closure of all windows and doors been rigorously enforced. In a modern air conditioned space with controlled air exchanges, temperature and humidity, little difficulty in obtaining a desired concentration would be expected.

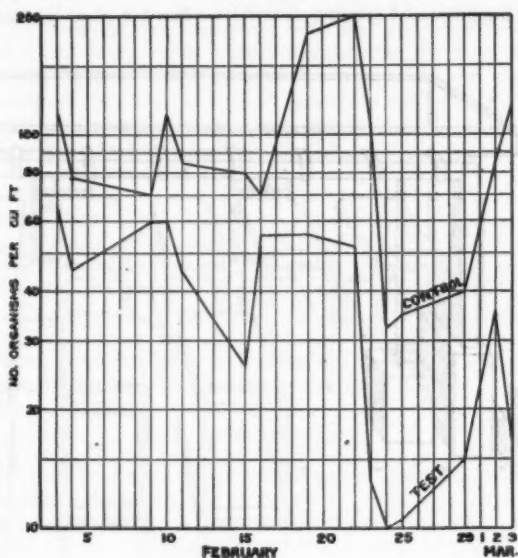


FIG. 9. BACTERIAL COUNT DATA—EACH POINT REPRESENTS AN AVERAGE OF TWELVE READINGS

An average relative humidity of 35 per cent was maintained in the test rooms during the test period. This value is in the optimum range previously mentioned.

The occupants of the test dormitories made no complaints regarding the odor of glycol vapors. Some were able to detect its presence in the room immediately when entering from the outside, but quickly became accustomed to it.

A marked reduction in the number of air borne bacteria was obtained in the test rooms. Results for one test period are shown in Fig. 9. There are several reasons why the spectacular reduction shown in Fig. 2 did not occur. As has been mentioned, the glycol concentrations were somewhat lower than desired. A more important reason is that the bacteria normally present in air include many resistant types as spore formers and various encapsulated organisms. In addition, it is believed many air borne bacteria are attached to dust particles, in which case sterilization may be more difficult.

Records were kept of the number of men from each dormitory contracting diseases believed to be air borne. Studies of the hospital records indicated a noticeable reduction of infections in the test dormitories.\* The results were very encouraging and further carefully controlled observations with larger groups are being planned.

#### CONCLUSIONS

1. Bactericidal quantities of glycol vapors can be maintained in large occupied spaces for long periods of time.
2. It has been shown that glycol vapors can be uniformly distributed in large rooms.
3. A means for generating glycol vapors in controlled amounts is described.
4. The occupants of treated rooms experience no discomfort due to glycol vapors.
5. Triethylene glycol is preferred over propylene glycol because much smaller amounts of the former are bactericidal.
6. Optimum bactericidal concentrations required with triethylene glycol range from 0.003 to 0.005 mg per liter of air (0.0013 to 0.0021 grains per cu ft).
7. Relative humidities in the range 25 to 60 per cent are required for optimum bactericidal action.

#### ACKNOWLEDGMENTS

A. F. Hubbard and R. H. Nelson, Herman Nelson Corp., Moline, Ill., kindly loaned the four fan units used in the field test and provided technical advice, E. DeCamp and M. Terrill, Philip Carey Co., Lockland, Ohio, gave much of their own time and effort in installing the ducts used in the laboratory and field tests. J. E. Kearns, General Electric Co., assisted in the design and manufacture of our improved vaporizer. This paper owes much to the technical assistance of Margaret Mellody and Silva Trautman, who carried out the bacteriologic and chemical work.

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\* A complete analysis of these data and the medical aspects of these tests is to be published elsewhere.

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## DISCUSSION

L. T. AVERY, Cleveland, Ohio: I would like to ask Professor Jennings, if any control was run concurrent with the glycol spray using only the humidity control; in other words, one control for a non-ventilated normally dry room, one with the humidity control and one with the glycol.

PROFESSOR JENNINGS: No, it was not, but we intend to do that in our tests which will start this year. Actually, we hope, in some of our spaces to run these vaporizers, some without any glycol under exactly the same conditions that would otherwise exist using glycol.



**1260**

## AIR DISINFECTION IN VENTILATION

By W. F. WELLS,\* PHILADELPHIA, PA.

EVER SINCE cave men vented smoke from their fires, ventilation has been associated with heating. Though temperature control still largely dictates air confinement, the problem today is rather to vent air vitiated by occupants of enclosed spaces than products of combustion. Before the dogma of *contagia* was displaced by the germ theory of disease, chemists measured ventilation load by determining the equilibrium concentration of carbon dioxide expired by occupants, which defined dilution with fresh air.

When bacteriologists caught infected droplets expelled in talking, coughing and sneezing on culture plates only within the immediate vicinity of a subject, they concluded that the flight range of infection could not exceed an arm's length and was therefore of little concern in ventilation. Most organisms in dust of inhabited spaces were found to be harmless varieties from decomposing matter rather than pathogenic organisms from diseased tissues. Chapin voiced the opinion, held by sanitarians, that contagion was spread indoors by *contact* between persons sharing enclosed atmospheres:

"Bacteriology teaches that former ideas in regard to the manner in which diseases may be air-borne are entirely erroneous; that most diseases are not likely to be dust-borne, and they are spray-borne only for two or three feet, a phenomenon which after all resembles contact infection more than it does aerial infection as ordinarily understood. Tuberculosis is more likely to be air-borne than is any other common disease." (1)<sup>1</sup>

Relieved of responsibility for venting air-borne agents of infection, engineers turned attention to physiologic effects of ventilation upon health. The regulation of temperature, humidity and air motion provided ample physiologic grounds for controlling the indoor climate within which civilized man spends ever more time, and ventilating engineers contributed air conditioning to the development of scientific control of our environment during the present century. Though hygiene resisted efforts to lower standards of ventilation, economy again pressed for further confinement of the air we breathe.

The environmental control of epidemic respiratory contagion by sanitary measures during this period did not match the reduction of intestinal infection conveyed by water, food and milk, or of insect-borne infection; nor did the strict application of aseptic techniques entirely eliminate hospital cross-infection. Epidemiology, focussed on residual respiratory infections, gradually dispelled the notion that the dynamic spread of contagion among aggregations

\* Associate Professor in Research in Air-borne Infection, Laboratories for the Study of Air-borne Infection (Supported by a grant from the Commonwealth Fund to the University of Pennsylvania for the study of the prevention and control of air-borne infection), Department of Preventive Medicine and Public Health, University of Pennsylvania School of Medicine. Member of A.S.H.V.E.

<sup>1</sup> Numerals in parentheses refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Grand Rapids, Mich., June 1944.

breathing common atmospheres could be wholly explained by a literal interpretation of *contact* infection. (2).

The classic work of Laidlaw and his associates, upon air-borne spread over considerable distance, of the viruses of dog distemper and influenza (3), Lurie's experiments on tuberculosis, dust-borne between animals in separate cages (4), observations on air-borne surgical infection by Hunt and Meleney (5), Cruickshank's studies of air-borne infection of burns (6), the Colebrooks' analysis of the role of nasopharyngeal organisms in puerperal infection (7), and the work of Allison on streptococcal infection in fever wards (8), and of McKhann on nosocomial infection in children's wards (9), are examples of the variety of evidence accumulated in recent years.

Improved bacteriologic procedures in sanitary air analysis also have interpreted Flugge's theory of droplet infection (10), proved quantitative inhalation of droplet nuclei infection to the lung (11), demonstrated habitual exchange during the winter months of respiratory flora among aggregations occupying enclosed atmospheres (12), and measured the sanitary inadequacy of present ventilation practice and the potentiality of air disinfection in control of dynamic spread of air-borne infection (13).

Elements of three types of *Bacteriologic Procedures in Sanitary Air Analysis* presented to the Committee on Ventilation and Atmospheric Pollution of the Industrial Hygiene Section of the *American Public Health Association* by the Subcommittee on Bacteriologic Procedure (14) have been summarized with special reference to air disinfection in the *Journal of Bacteriology* (15).

1. *Sanitary survey of inhabited atmospheres:* Rapid methods of collecting particles from large volumes of air enable the routine sampling necessary to obtain statistically significant indices of sanitary ventilation.

2. *Experimental studies of bacteria suspended in controlled atmospheres:* The mechanics of air-borne infection and control by ventilation have been studied bacteriologically in experimentally controlled atmospheres.

3. *Measurement of sanitary ventilation:* By quantitative sampling of test organisms added to atmospheres under different ventilating conditions, the hygienic importance of air disinfection has been demonstrated.

These procedures in 1935 disclosed higher bactericidal power of ultraviolet light against microorganisms suspended in dry air than in drinking water (16) and in the following year these observations were practically applied in surgical air asepsis (17). The effect of radiant disinfection of air in reducing surgical infection (18) and cross-infection in pediatric wards (19), a nursery (20), and an orphanage (21), and in the environmental control of epidemic spread of contagion in schools (22), now provides experimental evidence of the importance of air-borne infection.

While prevention of cross-infection in hospitals is only one field of ventilation, purveyors of breathing air in public places should also be concerned in the epidemic spread of air-borne infection. To make a public air supply proof against dynamic spread of contagion requires only that the number of new cases does not exceed the number of exposures, for then an epidemic can not grow. The epidemic spread of *volatile* childhood contagions in day schools has thus been checked when air disinfection eliminated test organisms ten times more rapidly than normal winter ventilation (23), and there is no reason to believe the epidemic spread of other air-borne infections cannot be controlled equally well by sanitary ventilation (24).

A Committee on the Study of the Use of Ultraviolet Rays as a Sterilization

Agent in Hospitals, of the Council on Hospital Planning and Plant Operation of the *American Hospital Association* reported favorably in 1940 (Bulletin No. 203). In 1942 the Council on Physical Therapy of the *American Medical Association*, after thorough investigation of radiant disinfection of air, issued a *Report on Acceptance of Ultraviolet Lamps for Disinfecting Purposes* (25). A conference of representatives of the Council, U. S. Public Health Service, *American Hospital Association*, the University of Pennsylvania, and the General Electric, Westinghouse and Hanovia Chemical and Manufacturing Companies met in Philadelphia at the time of the Annual Meeting of the A.S.H.V.E. to consider the problem of dosage in the radiant disinfection of air. Suggestions made at this conference were utilized by the Council in preparing its revised report. (*Journal American Medical Association*, 122:503-4, 1943.)

It was the belief of those present that progress had been made toward the codification of sanitary ventilation, insofar as approval of ultraviolet lamps was concerned, and that it would be necessary to meet again at some later time to consider the problem of intensity of irradiation within habited spaces. The necessity for this was recognized by the Council on Physical Therapy in its reports, *The Council cannot undertake supervision or assume responsibility for the satisfactory performance of any particular installation.*

A second conference on sanitary ventilation, called at the time of the 1944 Annual Meeting of the A.S.H.V.E. in New York City on January 31, also included, in addition to members of the first conference, representatives of the Army, Navy, Departments of Health and Labor of the State of New York and the New York City Health Department and Northwestern University.

The main problems which faced the second conference may be illustrated by an analogy. To intercept aircraft, it is necessary to lay down a properly directed and sufficient amount of flak, using weapons of suitable design. The responsibility for the design and production of the weapons belongs to the Ordnance Department; that for their proper use, to the Artillery. Likewise, in the interception of air-borne bacteria, the Council on Physical Therapy has assumed the responsibility for passing on the weapon, the source of radiation. But here the responsibility for the efficiency with which the weapons are used, the direction and intensity of the ultraviolet rays, had not yet been assumed by any authoritative body although the need has become manifest.

A formula was proposed (26) for evaluating radiant disinfection of air. The distribution of irradiation within enclosed spaces depends upon the length of the rays, as determined by their position and direction. Average intensity of irradiation, which corresponds to the concentration of disinfectants in chemical disinfection, may be determined by summing the products of radiation (watts) times length of rays (feet) between the source and the point at which the rays disappear from the atmosphere, divided by the volume of the enclosed space (cubic feet). This value, divided by the maximum uniform intensity obtainable from the radiation in a cubic space of the same volume, might be called the efficiency of irradiation.

Maximum disinfection of an enclosed space would be achieved if average intensity of irradiation were uniformly distributed throughout the entire space. In practice, however, it may not be feasible to expose occupants to the intensity required to disinfect. The amount of disinfection throughout the room, for a given average intensity of irradiation, is dependent upon circulation of the bacteria from the occupied to unoccupied zones. The average disinfection actually achieved can be determined by bacteriological tests, and the fraction

that this represents of the maximum, which is theoretically obtainable, can be computed. This fraction might be called the efficiency of disinfection.

The principal purpose of bacteriological tests is to determine the average rate of removal of significant air-borne organisms from the room by sanitary ventilation. Under special circumstances, as in use of light curtains, protection may exceed the average disinfection as defined by these tests. In general, however, the concentration of light in regions away from the occupants will result in less removal of organisms en route from person to person than by average disinfection in the room. The efficiency of irradiation and disinfection described above may indicate the extent to which protection to the occupants differs from that indicated by the average disinfection determined by bacteriological tests. It has been found in simple cases, such as schoolrooms, that with low efficiency of irradiation and disinfection, higher average disinfection would be required to accomplish approximately equal protection against classroom spread of certain childhood contagions. It was therefore proposed that the hygienic rating of disinfection be raised as the indices of irradiation and disinfection efficiencies are increased by good design.

These procedures for determining dosage apply particularly to irradiation of organisms in minute particles which drift around in the air like cigarette smoke until they are breathed into the lung, vented from the room with fresh air, or destroyed by disinfection.

Some believed that these are the primary cause of the epidemic spread of air-borne contagion, and that ultraviolet light is peculiarly effective against this type of spread (27). Others maintained that dust is of primary importance as a vehicle and that since coarser dust particles do not drift around in the same manner as droplet nuclei but tend to be raised, settled, and raised again without being maintained in the atmosphere, they were less likely to pass through the irradiated zones in the upper parts of the room. It was further suggested that even if exposed in the irradiated zones, they could not be so readily disinfected.

Bacteriological evidence was adduced that certain organisms survive in dust, where they have been found in considerable numbers both in the air and on surfaces (28). There is epidemiological evidence of the spread through air of streptococcal infections of the nose and throat by dust, and in hospitals, where lint from the bedding of infected persons may accumulate, this could be a serious matter (29).

In a serious study of haemolytic streptococci in the dust of hospital wards, and their relationship to infection, on the other hand, no conclusive example of a hospital cross-infection conveyed by dust was noted (30). Also the typical pattern of epidemic spread of contagion among aggregations of well persons could not satisfy conditions arising from the hypothesis of dust-borne infection; the generations observable in epidemics of measles, mumps, etc., would, contrary to epidemiological patterns observed (31), become indistinct if the infection was diffused in accordance with the activities which raise dust. On the spread of epidemic meningitis through troop transports, Greenburg states, "so far as we could see the upper and lower bunks did not show any real difference which would cause us to believe that the field of infection was more concentrated in the lower areas of the room" (32). Moreover, the rapidity with which influenza spread in 1918 and the sharpness of the decline (33) is not compatible with the dust hypothesis. Nor has the correlation

between extremely dusty atmospheres and epidemic spread of disease impressed the epidemiologist (1).

The difficulties of applying this formula in determining dosage required for disinfecting dust may render it impractical for such use.

It was suggested for controlling spread of infection by dust that all the lights be installed everywhere that the traffic would bear. Economy would not be a consideration. Others pointed out that, unlike droplet nuclei, dust might be controlled better by other measures than disinfection. It was easily removed from recirculated air by filtration or by electrostatic precipitation, or from surfaces by proper janitorial care, or laid by sanitary measures which make for air cleanliness (34).

Some considered that both problems should be given serious thought; that certain diseases might be spread more readily by droplet nuclei (*i.e.*, measles) and others by dust, as dried sputum in the spread of tuberculosis). In fact, air disinfection might provide a useful means of experimentally distinguishing modes of spread of respiratory infection and replacing opinion with significant data.

Furthermore, it was pointed out that the conditions for installing lights would also complicate the formulations. In crowded barracks with two and three levels of beds, the conditions would not be the same as in a schoolroom; nor would they be similar in a ward where the patients are confined to their beds.

Problems other than insuring adequate disinfection in an experimental demonstration were considered. For instance, in some diseases infection may spread or be prevented without clinical manifestations of the effect or irradiation, and bacteriological techniques must sometimes be applied to reveal the pattern of spread of infection. Chronic diseases like tuberculosis are difficult to subject to experimental study among human beings.

The question of susceptibility of the individuals in the aggregation under study may be vital if, as in measles, immunity is conferred by the disease. Therefore, the study of school children will differ from that of adults, although patterns of spread in influenzal infections may be similar in adults and children. Multiple infections, as in colds, may hide protection in the particular aggregation being observed (22).

Above all, if the study is intended to produce significant evidence for or against protection, the design of the experiment must provide a proper setting. Some means of identifying infections contracted within an atmosphere, or coverage of atmospheres shared by an aggregation sufficient to prevent multiple infections, is necessary to demonstrate protection by air disinfection. For instance, the incubation period may define secondary incidence of measles, but to determine whether influenza can be prevented in a barrack, it may be necessary to eliminate the possibility of widespread exposure in mess halls, recreation rooms, etc., of the individuals selected for study.

The new theory of air-borne infection revived interest in fumigation, which Chapin (35) so effectively dismissed. Although intermittent disinfection remains as a vestige of terminal fumigation, the interception of organisms en route from person to person by concurrent disinfection really is a ventilation concept. Leonard Hill (36) first proved the possibility of disinfecting air with tolerable concentration of atomized hypochlorates in 1928 and 10 years later Masterman developed this method to meet a growing demand for air disinfection (37).

Trillat, the French bacteriologist, at about the same time advanced a theory of aerosol disinfection based upon the hypothesis that particles of disinfectant would unite with particles of infectant to form a higher concentration of the former than would be possible with the same amount in gaseous form (38). British workers, developing this theory by atomizing hexyl-resorcinol dissolved in various glycols, obtained effective results even with negligible traces of hexyl-resorcinol suspended in propylene glycol (39). A group of Chicago workers under Robertson, attempting with detergents to improve this aerosol, found that propylene glycol without the hexyl-resorcinol accomplished similar results (40) and in vapor form gave results equal or superior to glycol aerosols (41).

Glycols, unlike former fumigants, are not true bacterial poisons and do not obey the normal law of disinfection. Their action seems to depend *inter alia* upon some hygroscopic quality intimately associated with humidity (42). This may help to explain why their action is closely limited to saturation ranges and why glycols with higher boiling points, such as triethylene glycol (43), and dipropylene glycol (44), which reach saturation in smaller concentration, disinfect in much lesser concentration than propylene glycol.

A clinical test of each of these three chemical methods of disinfection, empirically applied, has been favorably reported (45). Ventilation problems are, however, involved in the regulation of small concentrations of disinfecting gases, or vapors near saturation (46). The evaluation of practical performance on a wide enough scale and over sufficient time would seem to be a desirable subject for a future conference on sanitary ventilation; to consider the factors in the control of air-borne infection by chemical disinfection.

### CONCLUSION

Scientific investigations and practical experience during the past decade, therefore, indicate that air disinfection can reconcile growing pressure to cut the volume of fresh air displacement in air conditioning with growing demand for increased sanitary ventilation in the prevention and control of air-borne infection.

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## DISCUSSION

C.-E. A. WINSLOW, New Haven, Conn.: This whole development in what is now called sanitary ventilation is due chiefly to Professor Wells' work, which is one of the most significant things that has happened in public health in the last quarter of a century. It does not seem to me, however, that it is necessary any longer to debate very much the reality of the problem, the danger of air-borne infection, nor to argue about the index which has been found suitable for measuring the degree of that hazard; that is, the presence of this particular type of hemolytic streptococcus, to which Professor Wells refers.

It seems to me it is time to go on now and put this thing on a practical design basis. As a member of the Committee on Research, I personally hope that the TAC on Air Sterilization and Odor Control, of which Professor Wells is chairman, will attempt to do six things although they are difficult, or perhaps impossible.

What we need first is agreement on the method of determining the number of the streptococci present in air, and I think we should have fairly prompt agreement on that point. You know how difficult it is to secure standardization of instruments. You get an instrument that is 90 per cent accurate and then somebody comes along with an instrument or a method that is 92 per cent accurate, and you get a long delay in argument. It is more important just now to agree on a method which is reasonably accurate than it is to continue to squabble about minor differences in procedure. We want a standard method.

I hope, in the second place, that the committee will have the courage—and it will take some courage—to formulate a standard of attainment of performance, to state the number of these streptococci that can be present per cubic foot of air. That again will not be a 100-per-cent standard, but we can only get ideal standards by formulating an approximate one in the first place.

In the third place, we want statements from that committee to indicate the actual concentration of the streptococci as observed in various types of occupied spaces, not only on the average, but under extreme conditions. The engineer must deal with extremes of temperatures. He should also be able to deal with the extreme sanitary condition that occurs in an auditorium during the time of a cold epidemic.

Fourth, with these data the committee should be able to tell us how much air change is necessary to reduce various degrees of pollution to the standard by means of air change either with pure outside air or with recirculated air that has been adequately purified.

Fifth, we want to know how much it would cost to produce the same end result, the same performance standard, by the use of ultra-violet light or by a combination of ultra-violet light and air change.

Sixth, we want to know what it costs to obtain that result by the use of the aerosols that are sprayed into the air.

It is the quantitative economic efficiency of these three methods, air change, ultra-

violet radiation and spraying with aerosols that the engineer needs to know, in order to solve this problem intelligently.

W. A. DANIELSON, Memphis, Tenn.: It was in the spring of 1937 when I got to know what I call the two modern Curiés, Dr. and Professor Wells. I think for this meeting to pass without giving Dr. Wells considerable credit would be showing a lack of proper respect.

There are two points I would like to bring out for the purpose of obtaining a little more information. We installed a number of those sterilizing units, in ducts and directly over the operating table. In the ducts we did not make provision for cleaning the lamps and we got into endless trouble with that. Wherever you place lamps be sure to provide for easy cleaning.

Secondly, because tuberculosis is prevalent throughout the country, I would like to know a little more in detail about what has been done in trying to fight tuberculosis by this method.

AUTHOR'S CLOSURE: I wish to thank Dr. Winslow and General Danielson for their kind remarks and generous support from the beginning of our studies. We have always sought advice from Dr. Winslow, as the outstanding authority on air bacteriology; and we owe to his broad experience any success we may have attained. Never was his advice more welcome than now when our committee is getting under way; and with his guidance I am confident it will fulfill the tasks he has so clearly laid out.

General Danielson raises two vital points. The eye can observe dimming of electric lights due to dirty bulbs, but an electric eye is required to determine the loss of invisible ultra-violet radiation due to almost imperceptible films of dust or the oil from fingerprints. Lights usually are installed where they cannot be seen from occupied portions of the room and it is easy to overlook the servicing essential to effective disinfection. A distinct outbreak involving two classrooms called our attention to a change, without our knowledge, in janitorial arrangements; and inspection showed a heavy layer of dust which prevented the installation from performing its useful function.

Tuberculosis is of course one of the most important air-borne infections, and the waxy coat on the tubercle bacillus serves as protection against most chemical disinfectants. Our experiments have shown, however, that the relative vulnerability to ultra-violet light of pathogenic microorganisms, including the tubercle bacillus, is quite as uniform as the thermal death point in pasteurization. These experiments have been corroborated under more natural conditions by the prevention of animal cross-infection by the installation of ultra-violet lights between cages. This two-year study has just been reported in the *Journal of Experimental Medicine*. We hesitated to excite the medical profession or the public with false promises of security until this question had been thoroughly studied in the Laboratory, but we now believe these tests warrant full scale demonstration in sanatoria and offer additional reasons for air disinfection in public places, just as the destruction of the tubercle bacillus was one of the important reasons for pasteurizing milk.



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## SOME EFFECTS OF ATTIC FAN OPERATION ON COMFORT

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This paper is the result of research sponsored jointly by the Atlanta Chapter and the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station at the Georgia School of Technology.

TO MEASURE the effect of attic fan operation on comfort, an investigation was started by the Atlanta Chapter and the Committee on Research of the Society in cooperation with the State Engineering Experiment Station at the Georgia School of Technology. In a former report the results obtained in the summer of 1941 were presented to the Society.<sup>1</sup>

The data gathered came from tests conducted in two practically identical houses, one of which was equipped with an attic fan. Findings were reported on the effect of various air changes per hour on air temperature. Wet-bulb temperatures were noted and a few measurements of air velocity were made, but these data were so meager that they were not included in the report. It was decided to extend the investigation during the summer of 1942; the additional results are reported here.

### TEST EQUIPMENT AND INSTRUMENTS

#### *The House*

The two houses used during the summer of 1941 were not available for further use; consequently another Atlanta house was selected. It was a single story frame structure without a basement. The windows were double-hung wood sash, without awnings. The interior walls and ceilings were of wood, lath, and plaster. Fig. 1 is a floor plan of this house, showing the location of test instruments.

#### *The Fan*

The fan used was a four bladed axial or propeller type, 35 in. in diameter. It was rated to deliver 10,000 cfm at a speed of 350 rpm when operating against a static pressure of 0.01 in. of water. This fan was installed in the house in such a manner as to draw air through an opening in the ceiling of Room IV. The fan discharged air into the attic space. The gross area of

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<sup>1</sup> Exponent numerals refer to Bibliography.

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the opening in the ceiling was 17.5 sq ft and was covered by an expanded metal grille having 70 per cent free area. The net areas of opening by which air left the attic totaled 25 sq ft.

### Pyrometer

The air temperatures were measured with iron-constantan thermocouples. Because of the limited personnel, the thermocouples were attached to a sensitive potentiometer pyrometer of the automatic recording type, in order to study the complete daily temperature cycles. All air temperature measurements were made 30 in. above the floor level. The points at which temperatures were

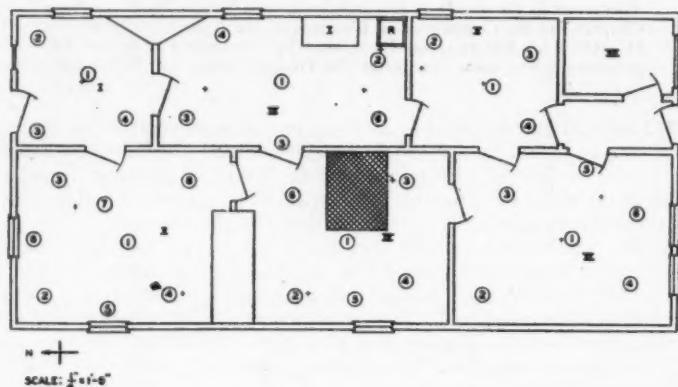


FIG. 1. FLOOR PLAN, SHOWING LOCATION OF INSTRUMENTS

+ AIR TEMPERATURE MEASURING STATION  
O AIR VELOCITY MEASURING STATION

I INSTRUMENT TABLE  
R RECORDING PYROMETER

observed are shown in Fig. 1 by means of cross (+) marks. In addition, temperatures of outside air and of air in the attic space were recorded.

Wet- and dry-bulb temperatures of the inside air for determination of relative humidity were recorded on a separate instrument.

### Anemometers

Velocities of air at points such as open windows were measured with a calibrated 4-in. anemometer. For studying air velocities at other points throughout the house *heated thermocouple anemometers*<sup>2</sup> of the type developed by Kratz, Hershey, and Engdahl were used. Recently a calibration was made for this anemometer, covering a range of 16 fpm to 3000 fpm was supplied by Professor Kratz.<sup>3</sup> Four of these anemometers were connected to a control panel with sufficient lead wire to reach the stations where velocity determinations were desired. The location of stations where air velocity was measured is shown in Fig. 1; each station is indicated by a circle. The velocities were determined at the same level as the temperatures—30 in. above the floor.

### *Globe Thermometer*

A globe thermometer was used to estimate the mean radiant temperature. The globe was constructed from a 6-in. copper sphere. This instrument was similar to the Vernon globe thermometer,<sup>3</sup> but the globe temperature was measured by a thermocouple as suggested by Houghten, Gunst, and Suciu.<sup>4</sup> The globe was supported 30 in. above the floor and was located in the center of the room. Further information on the use of this type of instrument has been reported by Bedford and Warner.<sup>5</sup>

### TEST METHODS

The fan was turned on during afternoons at times ranging from 3 to 5 p.m. and was turned off at approximately 9 a.m. the following morning. (Had a time clock been available the fan would have been turned off somewhat earlier in the morning.) During periods when the fan was operating the windows were kept open and the air temperatures were obtained from the charts of the recording pyrometer.

The procedure used for obtaining mean radiant temperatures was to place the globe in the center of the room 30 in. above the floor and to record temperature inside the globe, air temperature outside the globe not more than one foot from the globe. Air velocity at the same level as the globe, and not more than one foot from it, was recorded. Most of these measurements were made with the fan in operation, since it was desired to determine the effect of fan operation on mean radiant temperature.

The air movement and distribution was studied by using two or three procedures, each of which was somewhat similar to the others. One procedure was to put the fan in operation and determine average velocities at open doors and windows with the four-inch anemometer. The averages were obtained by the usual method of dividing the area into stations, taking one-minute readings of anemometer at each station, then adding the velocities, and dividing by the number of stations to obtain the average value.

The second procedure was to have the fan in operation and determine air velocity by means of a heated thermocouple anemometer at each station indicated on Fig. 1. During these tests the windows and doors of all rooms were opened.

The third procedure was the same as the second except that certain rooms were closed and others left open. The air velocity was then measured at all the stations in the rooms that were left open. In this procedure there were several possible combinations of rooms that could be left open. Obviously, Room IV, in which the fan grille was located, had to remain open in all cases. The following combinations were studied:

Rooms II and IV open; Rooms IV and VI open; Rooms III and IV open; Rooms II, IV, and VI open; Rooms III, IV, and V open; Rooms IV, V, and VI open; and Rooms I, III, and IV open. Air velocities were measured with the heated thermocouple anemometers.

### RESULTS AND DISCUSSION

Air temperatures were recorded on 15 days. The curves shown in Fig. 2 are typical of the results obtained. These curves show the variation of both inside and outside air temperature over a 24-hour period. During this test

the capacity of the fan was 45 air changes per hour based on the total volume of the house; all rooms were open. These curves confirm the results previously obtained in the earlier investigation.<sup>1</sup> It will be noted that while the fan was operating the difference between the inside and outside air temperature did not exceed three degrees Fahrenheit. Most of the time the temperature differential was not over two degrees Fahrenheit.

The results of the tests in which air movement and distribution were studied are shown in Tables 1, 2, and 3. Although the fan intake grille was located near the center of the house as shown in Fig. 1, there were large variations in air velocities and in quantities of air entering the several windows of the house. Table 1 shows the average velocity at each door and window when the

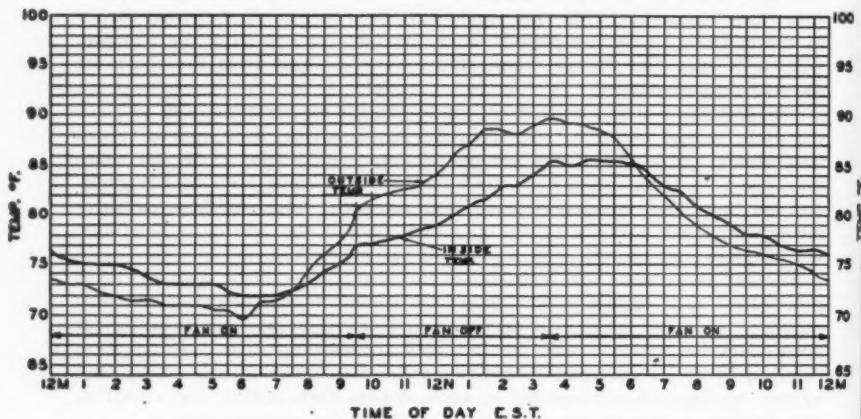


FIG. 2. RELATION BETWEEN THE TIME AND INSIDE AND OUTSIDE TEMPERATURES.  
45 AIR CHANGES PER HOUR 9-18-42

fan was in operation. It may be seen that in Room II the air velocity was 63 fpm at the north window, but was only 20 fpm at the west window. It is also interesting to note that the highest velocity was not obtained at the window of Room IV, the room in which the fan grille was located, but at the window of Room III.

The distribution of air in terms of cubic feet per minute and air changes per hour is given in Table 2. These quantities are based on air actually entering the rooms from the outside and do not include any air that may have passed through one room and then into another. The largest volume of air entered through the window of Room III, while the next largest entered through Room I. From this table it can be seen that the air changes per hour varied from 20 for Room II to 107 for Room I.

The differences in air volumes and air changes per hour were due, in part, to the resistance to air flow offered by various rooms. In general those rooms farther from the fan inlet would offer greater resistance and consequently have less air flow through them. Another factor that influenced the air

TABLE 1—AVERAGE VELOCITIES AT WINDOWS AND DOORS

	Room No.						
	I	II N	II W	III	IV	V	VI
Velocity at windows, ft/min. ....	144	63	20	291	188	48	67

Front outside door—30 ft/min.

Back outside door—22 ft/min.

Fan on—all windows and doors open

N-North

W-West

entering various rooms was, that a hill directly west and to the south of the test house sheltered the windows of Rooms II and IV from wind coming from that direction. The windows of Rooms I, III, and V were more favorably exposed to wind since the space east of the house was vacant; the nearest building was 200 ft away. A house located 40 ft south of the test house affected somewhat the amount of air entering the window of Room VI. In comparing air changes per hour the volumes of the rooms should be kept in mind. Since small rooms often have windows as large as the other rooms, the volume of air entering a small room may equal that entering a larger room; but, in such a case, the smaller room has a larger number of air changes per hour due to its smaller volume.

A careful check of the temperature data was made to ascertain if the large difference in air changes made an appreciable difference in temperature. It was found that the difference in temperature between rooms never exceeded one degree Fahrenheit.

The air velocities existing at various points in the house are shown in Table 3. With the fan in operation and all windows open, velocities from 18 fpm to 169 fpm were produced. A check of Table 3 against Fig. 1 will reveal the fact that most of the lower velocities were at points such as corners, where a rather small degree of air movement would be anticipated. It is important to note that while some locations had velocities much below the average, there was sufficient turbulence in the air flow to produce a reasonable degree of air movement even in corner locations. Table 3 shows also that the higher velocities occurred in rooms having greater number of air changes per hour.

TABLE 2—DISTRIBUTION OF AIR FLOW

Room No.	VOLUME OF ROOM CU FT	AIR FLOW CFM	AIR FLOW PER CENT TOTAL	AIR CHANGES PER HOUR
I	894	1595	22.8	107
II	1904	635	9.1	20
III	1664	1965	28.2	71
IV	1904	1440	20.6	45
V	954	605	8.7	38
VI	2064	745	10.6	27
TOTAL	9384	6985	100.0	45 Avg.

Above values based on air entering from outside.

TABLE 3—AIR VELOCITIES FEET PER MINUTE\*

ROOMS OPEN	Room No.																	
	I						II						III					
	Station No.						Station No.						Station No.					
	1	2	3	4	1	2	3	4	5	6	7	8	1	2	3	4	5	6
All.....	169	42	112	56	45	23	18	27	73	88	52	45	156	37	59	89	77	27
II-IV.....	..	..	..	..	158	16	18	47	166	145	82	82	..	..	..	..	..	..
IV-VI.....	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
II-IV.....	..	..	..	..	..	..	..	..	..	..	..	..	30	..	..	44	240	162
II-IV-VI.....	..	..	..	..	144	16	27	43	122	97	61	61	..	..	..	..	..	37
II-IV-V.....	..	..	..	..	..	..	..	..	..	..	..	..	122	52	159	317	243	37
IV-V-VI.....	..	..	..	..	..	..	..	..	..	..	..	..	..	36	266	79	165	36
I-III-IV.....	..	..	..	..	..	..	..	..	..	..	..	..	51	..	..	..	..	..

TABLE 3—AIR VELOCITIES FEET PER MINUTE—Continued

ROOMS OPEN	Room No.																	
	IV						V						VI					
	Station No.						Station No.						Station No.					
	1	2	3	4	5	6	1	2	3	4	1	2	3	4	5	6		
All.....	104	81	104	68	104	104	44	76	19	45	45	43	43	28	52	36		
II-IV.....	228	28	156	188	107	..	..	..	..	..	..	..	..	..	..	..		
IV-VI.....	354	57	210	28	210	28	..	..	..	..	..	..	157	84	41	157	66	240
II-IV.....	88	34	35	35	226	51	..	..	..	..	..	..	124	35	35	70	125	125
II-IV-VI.....	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
II-IV-V.....	57	75	41	46	105	181	118	81	39	65	..	..	..	..	..	..	..	..
IV-V-VI.....	167	85	300	85	230	164	81	115	43	141	32	40	81	32	46	32		
I-III-IV.....	83	40	48	48	88	111	..	..	..	..	..	..	..	..	..	..	..	..

\* Velocities measured with heated thermocouple anemometers—30 in. above floor.

Table 3 shows in addition the increase in air velocity that occurs in certain rooms when the doors and windows of other rooms are closed. It may further be noted that certain combinations of rooms were more effective in increasing air velocity than others.

A few apparently erroneous velocities in Table 3 may be explained by pointing out that the closing of doors in the house places some of the velocity stations in corners, and it also forces the air movement to occur along a somewhat different path.

The mean radiant temperatures were computed from the observed data by means of the Bedford-Warner formula.<sup>5</sup> An examination of a large number

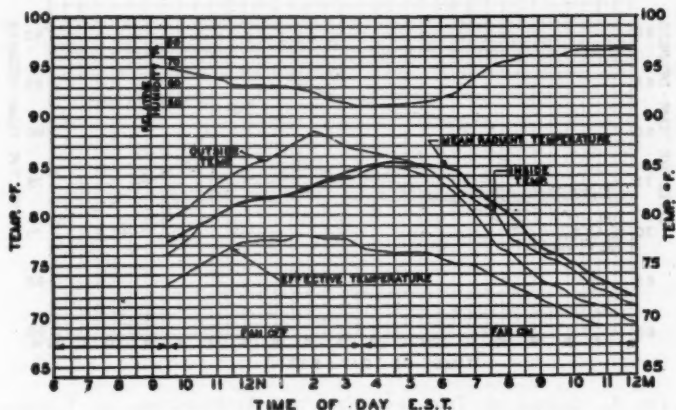


FIG. 3. CURVES OF TIME AND COMFORT FACTORS WITH 45 AIR CHANGES PER HOUR 9-11-42

of readings taken on five different days showed that mean radiant temperature was always very near room air temperature; the difference never exceeded three degrees. During certain periods of time, mean radiant temperature was equal to room air temperature. In the evenings when cool night air was being drawn through the house, mean radiant temperature was only slightly higher than air temperature. A great part of the time the difference was not over one degree. The results on mean radiant temperature are presented in the same manner as other temperatures. The curve of Fig. 3 is typical of the values observed.

This curve shows that attic fan operation reduces mean radiant temperature at approximately the same rate as it reduces room air temperature. The curve in Fig. 3 is for Room VI. This room was chosen because it was the southwest corner room and was expected to be affected by solar radiation during the afternoon to a greater extent than some of the other rooms because it was exposed to the sun in the afternoon.

An effective temperature curve for Room VI is shown in Fig. 3. This curve follows the same trend as the inside air temperature curve. It is quite

evident that when the fan was turned on the increase in air velocity produced an immediate drop of approximately one degree effective temperature.

An effective temperature of 73 deg is recommended<sup>6,7</sup> for summer comfort in the southeast portion of the United States when the dry-bulb temperature ranges from 91 deg to 95 deg. The effective temperature curve of Fig. 3 shows that this value of 73 was reached at 8:00 p.m. (E.S.T.) at which time the dry-bulb temperature was 78 deg and that it is reduced still further

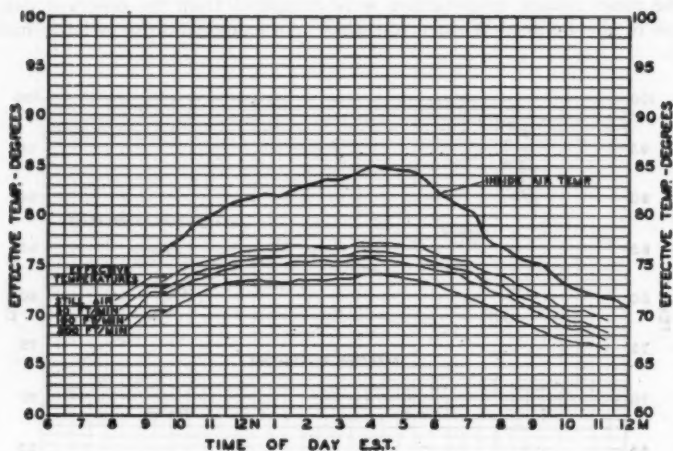


FIG. 4. CURVES OF TIME AND EFFECTIVE TEMPERATURES AT VARIOUS VELOCITIES 9-11-42

as the outside temperature decreases. (The values of effective temperature were obtained from a comfort chart.<sup>8</sup>)

In addition to the effective temperature curve of Fig. 3, the effective temperatures that would have existed in Room VI with various velocities have been plotted in Fig. 4.

#### DISCUSSION AND CONCLUSIONS

The mean radiant temperatures reported here are in agreement with what might be expected. During periods when the fan was off, transfer of heat between surfaces and air would tend to equalize the temperature of air and surfaces. During periods when the fan was on, the mean radiant temperature was only slightly above air temperature. Previous studies of night air cooling<sup>1,10</sup> have shown that in a frame house, surface temperatures are not more than a degree or two above room air. The surface temperatures reported have for the most part been ceiling surfaces, so the average would be less in some cases. Therefore, it would be impossible for mean radiant tem-

perature to exceed air temperature by more than one or two degrees in a frame house with this type of cooling.

It is possible that the mean radiant temperatures reported may not be the exact values for the time given, because of the time lag of the globe thermometer. The measurement of mean radiant temperature by the globe thermometer is based on the globe's coming into thermal equilibrium with its environment. In these tests the air temperatures and wall surface temperatures were continuously changing. Therefore, it is unlikely that true thermal equilibria were reached or that the mean radiant temperatures reported are exact. However, it seems reasonable to believe that the values given represent the magnitude and variation in mean radiant temperatures with good accuracy even though they may be slightly displaced as to time.

The results presented give some basis for a discussion of the proper number of air changes per hour. The attic fan used in a given installation should have sufficient capacity to produce the desired comfort conditions as early in the evening as possible.

From the viewpoint of air temperature, not more than 30 or 40 air changes per hour would be economical. Nevertheless, air velocity is a factor in comfort, and the selection of the number of air changes to use must take velocities into account. Fig. 4 shows the effective temperatures that would have existed in Room VI had the air attained the velocities indicated on the curves. When all rooms were open, Room VI had average of velocities of 41 fpm; thus it can be seen that comfort conditions of 73 deg ET were reached by 8 p.m. This room had 22 air changes per hour. For the same conditions Room III and 70 air changes per hour, and the velocities averaged 74 fpm, therefore, from Fig. 4 it is seen that Room III would have reached 73 deg ET at 7:30 p.m. or 30 min earlier. Using the same two rooms but selecting maximum rather than average velocities the results are: 73 deg ET reached by Room VI at 7:45 p.m. and by Room III at 6:45 p.m., or the comfort temperature is reached one hour earlier.

While these data indicate the desirability of high air velocities it is of interest to note a simple method of obtaining higher air velocities without the expense of an extra large fan. Table 3 reveals that when only Rooms IV and VI were open, Room VI had an average velocity of air of 124 fpm and would have reached 73 deg ET earlier than either of the previous illustrations.

The foregoing facts and illustrations indicate that while 30 to 40 air changes per hour are sufficient for reducing the inside air temperature to within two degrees of the outside temperature, it is desirable to produce higher air velocities so as to lower the effective temperature. The authors are of the opinion that satisfactory results may be obtained if fans are installed which are capable of producing 40 air changes per hour instead of the generally accepted standard of 60 air changes per hour. (These air changes are based on the total volume of the livable space in the house, and refer to actual air delivered and not to the maximum rated capacity of the fan.) At times when the outside air temperature is unusually high, the greater air velocities desired may be obtained by the simple expedient of closing the rooms which are not occupied. Attic fans are rated at free delivery. Flow resistances encountered in actual installations cause the actual air flow to be less than the rated capacity of the fan.

Measurements made at the intake grille show that the fan used in these tests delivered from 70 to 75 per cent of its rated capacity. Larger intake grilles or another type of grille might increase this percentage.

It is further suggested that means be provided to reduce air movement when comfort has been attained as an examination of Figs. 3 and 4 will show that what might be a pleasant breeze at 9 p.m. may become an objectionable draft by 12 p.m. unless the fan is stopped or its capacity reduced. Although a variable speed motor and thermostatic controls might be used to accomplish this, the most economical means is to use a time clock fan which operates a switch to stop the fan motor at a time selected by the occupants of the house.

It is interesting to estimate the difference in effective temperature between a house with attic fan and one without an attic fan. Tests have shown that air temperature in a frame house does not decrease much before 8 p.m. if no fan is used. Fig. 3 shows that 84 F dry-bulb may be assumed for air temperature in the house at 8 p.m. If the air is still, as it would be without a fan, an effective temperature of 80 deg results; this is seven degrees higher than the 73 deg ET attained at 8 p.m. with attic fan in operation.

The small difference between air temperature and mean radiant temperature indicates that this difference is not a large factor in determining fan capacity. Since mean radiant temperature will always be near air temperature it is important to observe that if no fan is used mean radiant temperature will be higher than with a fan and will contribute to the discomfort experienced.

The following is the principal conclusion that may be drawn from attic fan data available. The comfort produced depends primarily on only two things for a given house, the conditions of the outside air and the velocity that the fan produces.

#### SUMMARY OF RESULTS

The results of tests on a single story frame house equipped with an attic fan, and located in Atlanta, may be summarized as follows:

1. The inside air was found to be approximately two degrees Fahrenheit above the temperature of the outside air when average number of air changes per hour for the house was 45.
2. With an average for the house of 45 air changes per hour some rooms had as low as 20 air changes per hour while others had in excess of 100.
3. In spite of different rates of air change for various rooms, the air temperature did not vary more than one degree Fahrenheit from one room to another.
4. Large numbers of air changes are useful only because the increased air velocity decreases effective temperature.
5. The average mean radiant temperature was found to be about one degree above air temperature. The difference never exceeded three degrees.
6. With the fan in operation and all doors and windows open, air velocities varied from 18 to 169 fpm. The lower values were obtained in corners.
7. When some of the rooms were closed the velocity of air at points in other rooms showed a marked increase. Increases of 100 per cent were noted in many cases and 200 per cent in a few locations.
8. With the fan in operation during the evening effective temperature decreased at approximately the same rate as outside dry-bulb air temperature.

#### ACKNOWLEDGMENT

The authors wish to acknowledge and to express appreciation for valuable assistance to Prof. A. P. Kratz, University of Illinois, who supplied informa-

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#### DISCUSSION

T. T. TUCKER, Atlanta, Ga. (WRITTEN): The authors have again produced outstanding data in this *extended investigation*. These data possess a startling degree of practicality.

There is one procedure which perhaps should be explained. Why were readings taken 30 in. above the floor? Why were they not taken at some other elevation such as 60 in. above the floor?

From the authors' explanations of the differences noted in air changes and air velocities in various rooms, it would appear that best comfort results from attic fans can only be procured when effects of external surroundings are properly weighed. These must include: 1. Orientation of the building; 2. Location with respect to immediate geography; 3. Direction of prevailing winds; 4. The proximity and location of adjacent buildings. It would materially add to the usefulness of this paper if the authors would present us with succinct conclusions on these points.

It is a known fact that a well insulated house, with proper shading devices over the glass areas, will in the Atlanta area start its cooling cycle many degrees below that a house such as the test house described in this paper. It would appear, therefore, that the operation of an attic fan under such conditions would be entirely different from that shown by this paper. For example, the optimum time of day for starting and stopping the fan would certainly change. In view of the ever increasing use of good insulation and proper shading devices I wonder if the authors intend to investigate these in combination with the attic fan?

It can be seen from Fig. 2 that when the fan is first turned on it draws air into the house at a higher temperature than the air displaced. Would it be reasonable to assume that the initial cooling effect is procured by exhausting the hot air from the attic? I wonder if best results could be obtained by displacing the attic air only until such time as the temperature of the outside air had dropped below that of the inside air? Do the authors have any data on this method of operation?

C. F. MALLEY, Detroit, Mich.: More interest should be shown in this subject, particularly when fans are more available. One of the biggest items not mentioned is that when the humidity is rather high, as it is in Michigan, and certain rooms are closed, the aspirating or the entrained effect of the air coming through the windows under high velocity produces a cooling effect. If a velocity considerably above those mentioned in the paper is obtained, for example 200, 300 or even 400 fpm, the actual effect becomes not so comfortable but chilly and drafty. However, the entrainment effect of that air coming in under high velocity provides turbulence throughout the room and obtains a pleasant cooling effect. That has been proven by taking thermometer readings. When 45 to 100 fpm turbulent velocities throughout the entire room are produced the results obtained will be best.

One bad result of many air changes is an increase in dust. A certain amount of dust is always present in the air and it finds its way into the house.

In the case where relative humidity is 70 to 90 per cent, as in Detroit, the aspirating and consequent turbulence and cooling effect of this air is greatest, and since you do have a cooling effect it leads me to believe that in the Michigan area even evaporative type coolers which are normally not sold because of the high humidity they put into the air, would have some value. As long as air is less than saturated it still has some hygroscopic effect and it will evaporate that film of humid air near the skin to produce some comfort. I have a system in my home and office which gives a very fine effect in view of the fact that the solar exposure is very great and the temperatures quickly build up without the fan to 15 and 20 deg above the outside temperature. By increasing the air changes to as many as one in 30 seconds, which is 120 an hour, a very satisfactory type of cooling effect is obtained.

G. L. TUVE, Cleveland, Ohio: I would like to call attention to one item in connection with the work reported in this paper, and that is the matter of instrumentation. I noticed yesterday, in attending five different technical advisory committee meetings, that every one of them was talking about instrumentation.

The instrument used by the authors for air velocity measurements, the heated thermocouple anemometer, is a rather new instrument. It was developed as a

part of our cooperative research at the University of Illinois. Most of us do not know very much about it, and it may not give exactly the same readings as some of the other instruments we have been using. Many of us have been using the velometer. I think the original work on the comfort zone chart was done with Kata thermometers. While the Kata thermometer is a thermal instrument like the heated thermocouple, it probably has more lag. Effective temperatures measured by various instruments should be considered as approximations rather than as exact measurements, until we know more about our instruments.

B. H. JENNINGS, Evanston, Ill.: I would like to make two requests of the authors: one of them is to discuss the problem of dust deposition in this house with the large number of air changes and state about what dust conditions exist in that locality. The second is to have the authors give some information on the power required to run the attic fan.

AUTHORS' CLOSURE: I would like to thank all those who contributed to the discussion. I will try to answer some of the questions but do not think I can answer all of them. Mr. Tucker asked about the air temperatures and velocities being measured 30 in. from the floor. I had noticed in recent literature that the 30 in. level was being used because it corresponds more nearly to the mean height of the person seated. It is undoubtedly true that the environment and the orientation of the house influence the results. Rooms on the side from which the prevailing wind comes would be favored by having higher air velocities. It is not definite, but it seemed to be indicated rather strongly that in the house tested, adjoining buildings, on adjacent lots, influenced the air velocities. The house faced north. The west side of the house was somewhat sheltered by other houses, and by being at the foot of a hill, the hill being to the west of the house. That side of the house seemed to indicate somewhat lower air velocities. On the other side, the east side, was a vacant lot for 200 ft or more, and that seemed to give that side a favorable position. Now, as to what can be done about that, or what Mr. Tucker had in mind, I do not know. Further study might throw some light on that. It might be possible that an exposure factor could be used.

We did not study the effect of the attic fan when drawing air through the attic only. I think we can safely say that operation of the fan during the daytime for the purpose of moving air into and out of the attic, would increase the comfort because it is a well known fact that attic temperatures will reach 120 or 140 deg or more, if the attic is not ventilated. If the attic temperature is lower then less will be transferred into the house. I do not have any figures as to how much good that will do, but it certainly will help.

In connection with the statements that Professor Tuve made about the instruments, I would like to make two statements not only about the anemometers but about the other. The heated thermocouple anemometers that are used were obtained through the kind cooperation of Professor Kratz and his mechanic who made the sensitive measuring elements from the jigs and fixtures that he has, and they kindly supplied the calibration with it. We attached it to our own potentiometers and control panels. It is, of course, true that there are a number of types, as Professor Tuve points out, and some other instrument might give somewhat different values. I used that rather than the Kata thermometer, for the reason mentioned by Professor Tuve that the Kata thermometer is rather slow. We had four of them which we could read consecutively while sitting at the instrument table.

Professor Jennings asked about dust, and in answering I can say that it was bad. The first summer of test work was carried on in two houses that were owned by a cotton mill company and as there was considerable lint in the vicinity a lot of it caught on the screens and a lot of it passed through. I do not know just what can be done to keep out dust but I can say definitely that a lot of dust and trash enters at these very high air velocities in a very dusty location.

Power requirements there have been studied in a number of actual installations. Quite a lot of information on that is recorded in a Texas A & M Bulletin.<sup>†</sup> I can say that the fan we used, a 10,000 cubic foot fan, was operated by a one-third horse power motor.

<sup>†</sup> The Installation and Use of Attic Fans, by W. H. Badgett. (Bulletin, Agricultural and Mechanical College of Texas, College Station, Tex., April 1942, 45 pp.)



**1262**

# TRAIN PISTON ACTION VENTILATION AND ATMOSPHERIC CONDITIONS IN CHICAGO SUBWAYS

By WALTER E. RASMUS\* AND EDISON BROCK\*\*

## INTRODUCTION

THIS is the report of a survey of air movements resulting from train operation, and of atmospheric conditions in the State Street route of the Chicago subways, which was opened to the public in October, 1943.† The effectiveness of the piston action of trains in providing large volumes of tunnel ventilation has long been recognized. As early as 1910 air flow measurements resulting from the joint action of trains and exhaust fans were made in the tubes of the Hudson and Manhattan Railroad Co. and reported in connection with an investigation of ventilation.<sup>1</sup>

In 1938 when construction was started on the Chicago subways a review of available technical literature disclosed frequent general references to the subject but meager specific data and no comprehensive theoretical analysis. It was realized that the anticipated heavy traffic operating in single track tubes would cause severe piston effect upon the air and also give out great quantities of heat which would have to be disposed of. In order to coordinate available data so that it might be applicable to the Chicago subways and to establish standards for design of ventilation facilities, a method for calculating the effect of train piston action on the subway air was originated and this theoretical analysis was extended to include a heat balance.<sup>2</sup>

It was not until much of the work on the Chicago tubes was completed that the account of a ventilation survey relative to the problem of smoke removal in the Moffat Tunnel was released. In that report the analysis of train piston effect and accompanying formulae and air flow diagrams are supported by field measurements with both train and fan operation.<sup>3</sup>

It is acknowledged that no exact theoretical solution of air flow is possible in the case of the typical rapid transit subway with its numerous station entrances, vent shafts, open platforms and with trains varying in frequency, speed and direction of travel. For the same reasons it is equally true that no

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† Investigation conducted by the engineers of the Department of Subways and Superhighways, the Federal Works Agency and the Chicago Rapid Transit Co.

<sup>1</sup> Ventilation of the Hudson River Tubes, by A. W. Hodgson. (*Heating and Ventilating*, Vol. VII, No. 5, May, 1910.)

<sup>2</sup> Development of Formulae for Calculating Ventilation for the Chicago Subways, by Edison Brock. (*Journal of the Western Society of Engineers*, Vol. 48, No. 2, June, 1943.)

<sup>3</sup> Piston Effect of Trains in Tunnels, by R. L. Daugherty. (*A.S.M.E. Transactions*, Vol. 64, No. 2, February, 1942.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Grand Rapids, Mich., June, 1944.

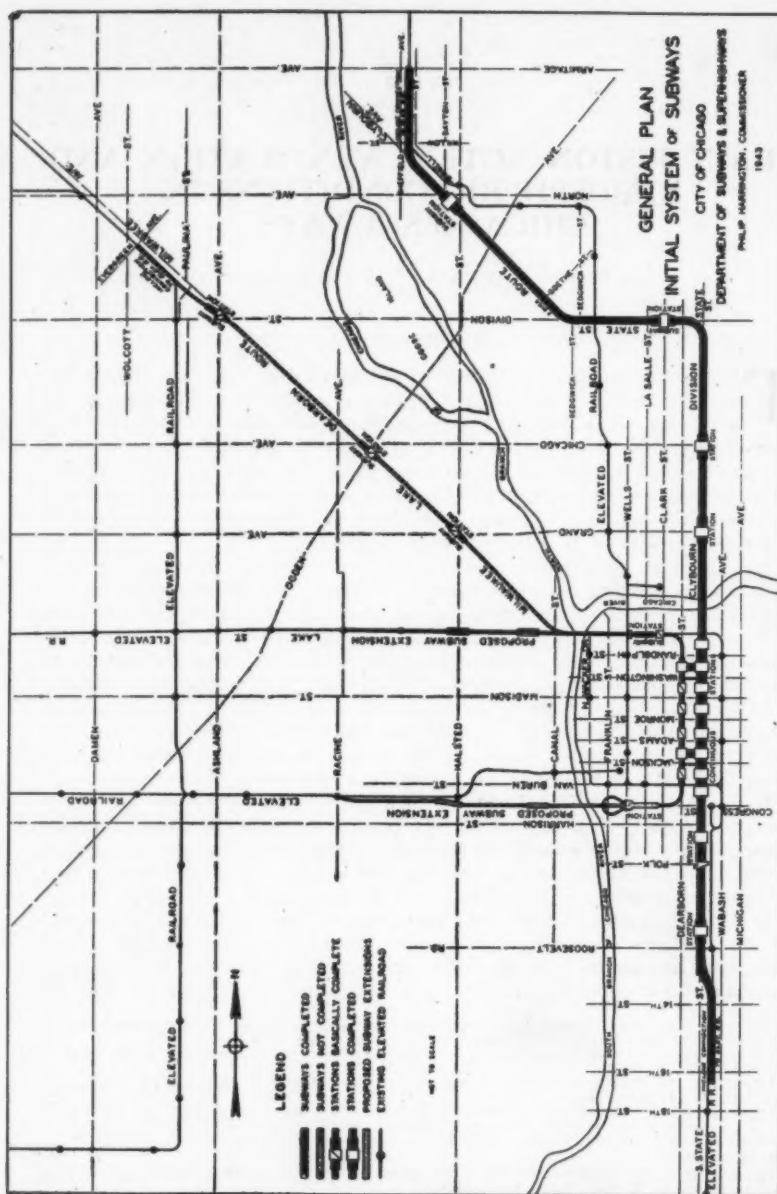


FIG. 1. GENERAL PLAN, INITIAL SYSTEM OF SUBWAYS

fully accurate determination of air flow by field measurements is possible. However, a theoretical analysis is of value in establishing design requirements and an operating survey should follow to determine if these objectives have been attained.

The survey described herein does not pretend to have laboratory exactness but it has been conducted as carefully as practical considerations permitted, and it is believed that the resultant data are amply accurate for the purposes intended.

#### DESCRIPTION OF CHICAGO SUBWAYS

The design of the Chicago subways with respect to ventilation has been previously described and illustrated,<sup>4</sup> and an engineering summary of the work

TABLE 1—STATE STREET SUBWAY DIMENSIONS

TYPE	LENGTH (FT)	AREA (SQ FT)	PERIMETER (FT)	VOLUME (CU FT)
<b>Portals:</b>				
South.....	568	225	60	127,800
North.....	700	266	67	186,200
Horse-shoe Section.....	24,271	235	64	5,703,685
Circular Shield Section.....	4,214	290	70	1,222,060
<b>Crossovers:</b>				
Roosevelt Road.....	450	687	119	309,150
Illinois Street.....	177.5	509	93	90,348
Clybourn Avenue.....	177.5	509	93	90,348
<b>Station Sections (Side Platform Type):</b>				
North Avenue.....	1,000	340	163	340,000
Chicago Avenue.....	1,000	340	163	340,000
Grand Avenue.....	1,000	340	163	340,000
<b>Station Sections (Center Platform Type):</b>				
Clark Street.....	506	721	130	364,826
Downtown.....	3,413	623	139	2,126,299
Harrison Street.....	625	623	139	389,375
Roosevelt Road.....	683	622	151	424,826
<b>TOTAL.....</b>				<b>12,054,917</b>

will be available to the reader.<sup>5</sup> The following description is therefore limited to the survey of train piston action ventilation and atmospheric conditions.

Fig. 1, showing a general plan of initial system of subways for Chicago, includes both State Street and Dearborn Street routes. The former, which was placed in service in October 1943, is connected at both portals by inclined tracks to the elevated lines of the Chicago Rapid Transit Co. and certain express services are now routed through the subway. Heavy construction work for the Dearborn Street subway has been completed for the greater portion of its length, but installation of station finish and equipment has been temporarily deferred because of the war. Both routes have two deep level tubes, each with a single track, and these subways are independent

<sup>4</sup> Ventilation of the New Chicago Subway, by Walter E. Rasmus. (*Heating, Piping & Air Conditioning*, Vol. 15, No. 8, August, 1943.)

<sup>5</sup> Construction of Chicago's First Subways. (A series of papers by 17 Engineers, presented in abstract January 6, 1944, to be published in the *Journal of the Western Society of Engineers*, Vol. 49, No. 2, Part 2, June, 1944.)

except for block-long pedestrian passageway connections, four in number, in the downtown area. While the construction and ventilation provisions are similar, the following description is confined to State Street subway.

North of the Chicago River the stations at Grand Avenue, Chicago Avenue and North Avenue are of the side platform type with the central wall of the tubes continued between tracks. The Clark Street station, and all stations south of the Chicago River, have open central platforms serving both tracks. All platforms are 500 ft long except that one platform 22 ft wide and 3,400 ft long serves the eight downtown stations. There are usually four sidewalk

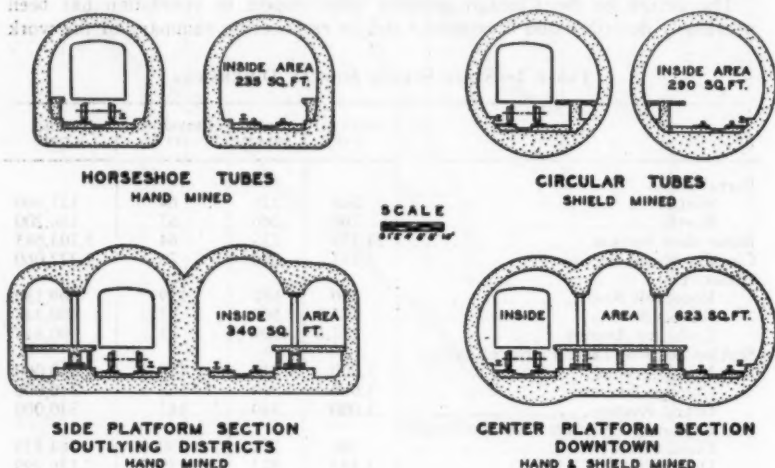


FIG. 2. TYPICAL SECTIONS OF STATE STREET SUBWAY

entrances and stairways between the street surface and each mezzanine station and two combinations of an escalator and a stairway connecting the mezzanine with the platform level. The essential dimensions of the State Street subway are shown in Table 1. Typical sections of State Street subway are shown in Fig. 2.

#### TEST PROCEDURE

In order to observe the piston action of trains upon the subway air under as nearly uniform conditions as possible, all air flow measurements were made during the off-peak period between 9:00 a.m. and 3:00 p.m. The average train consisted of four 50-ft all steel cars running on a three minute headway with approximately 40 trains per hour on the two tracks. Maximum train speeds were 29 mph to 38 mph, depending upon distance between station stops. During morning and evening rush hours, train lengths are increased to eight cars, but this is offset by lighter traffic after midnight, so that the observations represent a fair average under present traffic conditions.

Air velocities were measured with an Alnor Boyle type of velometer

having two scales giving instantaneous indications to a maximum of 500 fpm and 2,500 fpm, respectively. On account of fluctuations in velocity and frequent reversals in direction of air flow this type of instrument proved more suitable than an anemometer.

During each observation air velocity indications were recorded on a log sheet at 5-second intervals for a period of not less than 10 min. In order to secure a fair average, at least three observations were made at each location, preferably on different days. Such procedure was necessary because variation in train headway cannot be avoided and this, together with frequent station stops and the combinations of effects of trains operating on adjacent tracks, causes an irregular pumping action with peak velocities normally outward and then inward as each train passes. As the present survey under one set of conditions requires approximately 50,000 instantaneous meter readings, the average results should establish the characteristic subway air movement with reasonable accuracy.

Since the two train tubes are interconnected and the air circulates rapidly along the tracks between the subway sections, it was necessary to measure velocities in all structures and pedestrian passage-ways connecting the tubes to the outer atmosphere. The air flow into the downtown section by way of the tunnels was also measured to permit a summation of air movement in this section. The number and approximate area of openings from train tube level to the street surface are as follows:

LOCATIONS		Sq Ft
4	Train tube entrance at subway portals.....	982
32	Escalator and stairwells from train platforms to mezzanine stations.....	2,320
36	Standard blast shafts in tubes adjacent to stations.....	3,600
44	Standard vent shafts with louvers between stations.....	4,400
13	Downtown station vents with fans.....	482
TOTAL 129		Approximate area 11,784

The standard vent and blast shafts, which are similar in design, provide ventilation and balance air pressures, the latter located immediately in advance of and beyond station platforms to limit air velocities at these locations. Each vent or blast shaft unit is composed of two to four cylindrical vertical wells with a combined area of 100 sq ft, with connections at the base through the tube wall. Each terminates in 120 sq ft of sidewalk grating. As the survey reported herein was conducted in winter, the vent shaft louvers were closed to retain heat but the blast shafts were open. There are 26 fans with a combined capacity of 1,124,000 cfm for supplementary ventilation and emergency use. To obtain an accurate record of train piston action alone, none of the fans was operated during the test period. The fans are the axial flow type without housings. They are mounted in the air vents for economy and space saving, and the fan discharge ducts serve also for the passage of piston action air flow. The arrangement of fans and various structures for ventilation has been described and illustrated.<sup>6, 7</sup> The air flows measured

<sup>6</sup> Loc. Cit., see Note 4.

<sup>7</sup> Loc. Cit., see Note 5.

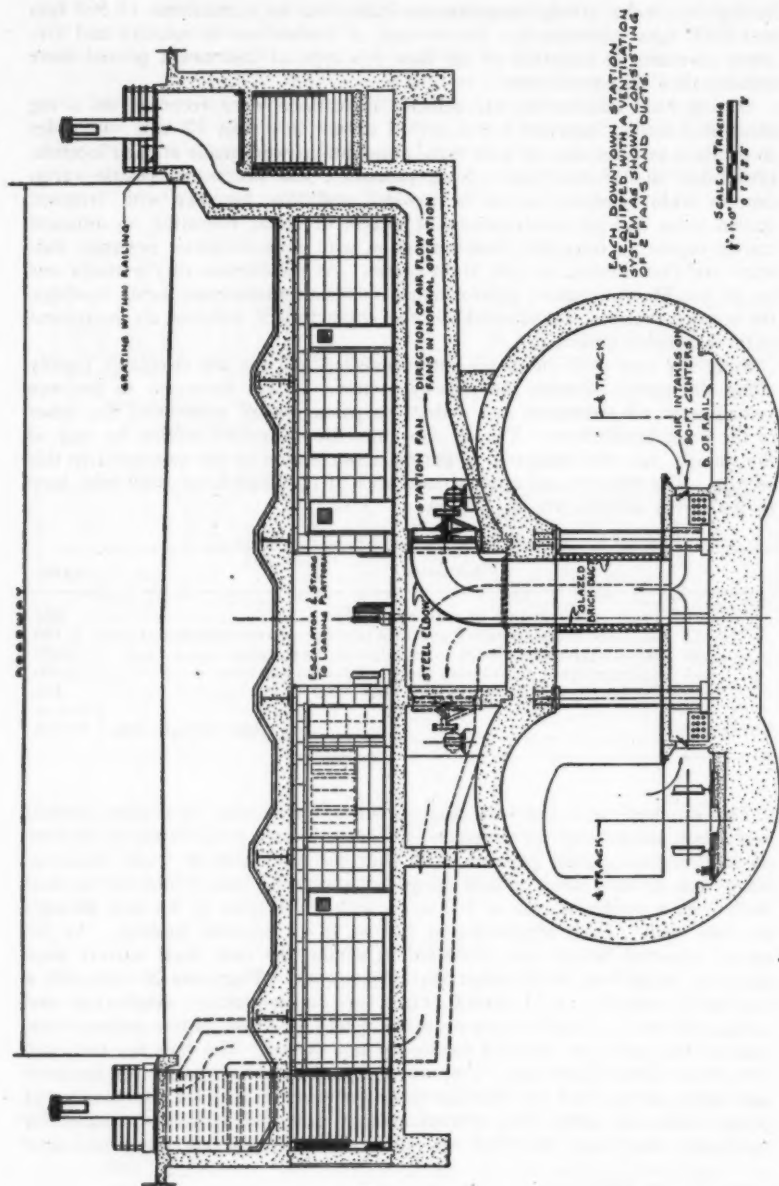


FIG. 3. TYPICAL VENTILATION SYSTEM FOR DOWNTOWN STATIONS

were due to train operation, natural influences being negligible. Ventilation can be increased at will by opening vent louvers and by operating the fans.

Inasmuch as one of the main functions of subway ventilation is the disposal of heat generated by car motors and other equipment, a continuous record has been kept of subway temperatures by means of 12 recording thermometers installed at representative locations. They are of the self contained type with 7-day manually wound clock movements, 0 F to 100 F scale, and are guaranteed to be accurate within one deg above or below the exact air temperature.

In addition to the general survey, a number of specific investigations were conducted, several of which are described in this paper.

#### RESULTS OF AIR FLOW TESTS IN DOWNTOWN STATION AREA

While the survey for the State Street subway has progressed sufficiently to warrant a general discussion of the characteristic air flow and atmospheric conditions for the entire subway, further meter readings will be required at certain locations to confirm the present data. However, three or more sets of observations have been made throughout the downtown section from Lake Street to Congress Street. The results are shown in Table 2 and Fig. 4.

The location of this area is indicated in Fig. 1 and a typical cross section is shown in Fig. 3. It will be noted that the parallel tubes, serving the northbound and southbound tracks, open to a central platform 22 ft wide and 3,400 ft long. This platform serves eight mezzanine passenger stations, each of which has four stair entrances in the sidewalk and two sets of a stairway and an escalator in combination connecting with the train platform, except that the Van Buren-Congress station, the southernmost of this group of stations, has but one such joint escalator and stairway. Beyond the ends of this continuous platform, the train tubes are separated, each with its single track.

Table 2 gives a summary of air movement in and out of the platform level of this section at all possible locations; that is, at the four train tube connections of which there are two at each platform end, at the 15 escalator and stairway combinations, and at the 13 fan ducts. In each downtown station, these fan ducts terminate adjacent to the southwest and northeast stair entrances. Since they serve as air vents irrespective of fan operation, they are designated as S.W. and N.E. air vents. The summation of air flow indicates 403,395 cfm supply and 413,229 cfm exhaust for this section of subway. The mean, 408,312 cfm, is considered as representing this section's characteristic air flow. This is sufficient to change the 2,126,300 cu ft volume of the section 11 times per hour. Since the fans were not being operated when this survey was made, and natural effects were found to be minor, this movement of air may be assumed to represent the average ventilation due to the piston action of the trains.

It is evident that all of this air supply does not come directly from out of doors but its volume is many times that normally supplied for breathing purposes, and it is comparatively fresh, much of it having entered the subway at nearby blast shafts in the train tubes.

Such a large volume of piston action air movement had not been anticipated. The survey also revealed a pronounced characteristic air flow of even greater value in obtaining and maintaining excellent atmospheric conditions in these heavily patronized downtown stations. It will be noted that the predominant

TABLE 2.—TABULATION OF TEST RESULTS OF DOWNTOWN SECTION OF STATE STREET SUBWAY

No. of Readings	Location of Observations	Velocity of Air Flow fpm				Duration of Air Flow per cent			Air Flow cfm	
		Max. Out	Max. In	Average Out	Average In	Out	In	None	Out	In
360	Lake-Randolph, N. Stair.....	900	400	233	140	64.2	30.3	5.5	12,865	3,659
360	Lake-Randolph, S. Stair.....	525	350	202	105	61.0	33.3	5.7	10,588	3,008
360	Randolph-Washington, N. Stair.....	700	470	256	201	72.5	20.3	7.2	15,916	3,494
360	Randolph-Washington, S. Stair.....	900	500	279	248	53.9	37.5	8.6	12,982	7,970
360	Washington-Madison, N. Stair.....	800	600	323	193	76.1	15.8	8.1	21,130	2,614
360	Washington-Madison, S. Stair.....	700	550	240	264	43.2	48.3	8.5	8,889	11,048
360	Madison-Monroe, S. Stair.....	700	500	368	248	76.3	18.3	5.4	24,173	3,909
360	Madison-Monroe, N. Stair.....	900	400	270	148	69.4	24.2	6.4	16,159	3,084
360	Monroe-Adams, N. Stair.....	800	350	240	76	80.0	18.7	1.3	16,548	1,235
360	Monroe-Adams, S. Stair.....	950	550	274	168	78.0	17.9	4.1	18,389	2,581
360	Adams-Jackson, N. Stair.....	500	550	282	232	62.8	33.9	3.3	15,192	6,720
360	Adams-Jackson, S. Stair.....	850	400	316	186	81.7	12.5	5.8	22,152	1,984
360	Jackson-Van Buren, N. Stair.....	850	500	252	200	70.6	24.4	5.0	15,298	4,230
360	Jackson-Van Buren, S. Stair.....	1,700	450	303	145	75.9	19.1	5.0	19,769	2,391
360	Van Buren-Congress, Stair.....	006	650	252	309	20.6	69.8	9.6	4,467	18,461
5,400	Total air flow via 15 combined stair and escalator wells.....								234,517	76,388
4,680	Total air flow via 13 air vents (Station fan discharge ducts).....								24,882	12,032
720	Total air flow via tubes at north end—North Bound Tube.....								69,215	3,545
	South Bound Tube.....								1,150	155,213
720	Total air flow via tubes at south end—North Bound Tube.....								77	152,598
	South Bound Tube.....								83,888	3,619
11,520	Total air flow in downtown station area.....								413,229	403,395

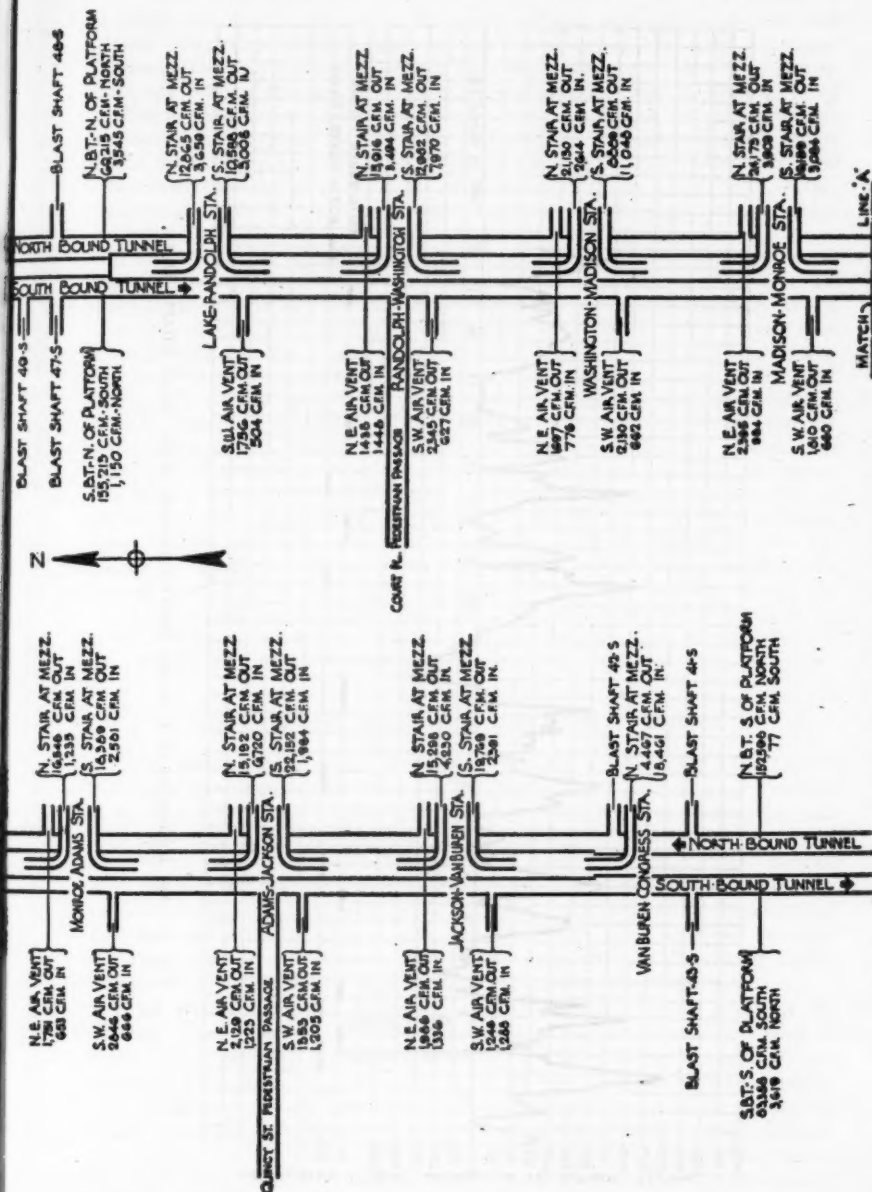


FIG. 4. CHART SHOWING RESULTS OF AIR FLOW MEASUREMENTS IN DOWNTOWN SECTION OF STATE STREET SUBWAY

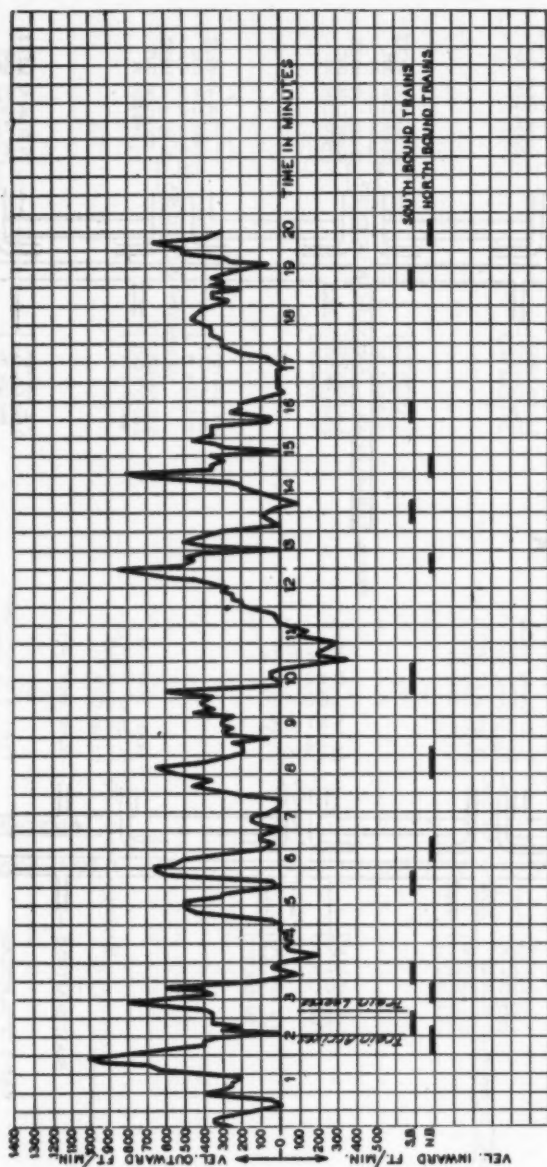


FIG. 5. AIR VELOCITIES IN CHICAGO AVENUE STATION WITH BLAST SHAFTS OPEN.

combined air flow is inward at the two-train tube connections at each end of the continuous platform and outward at seven of the eight passenger stations. The only exception is at the least important southernmost Van Buren-Congress Station through which the predominant air flow is inward, being similar to that through the adjacent tubes. As a result of this characteristic air movement, tempered air averaging 40 F even in the coldest weather, enters at the train tube connections and by mixture and heat from walls, trains and other sources, quickly rises to an even temperature of about 50 F. This temperature generally prevails at track level and in the important mezzanine stations as well as in the stairways to the street surface.

A combination of conditions account for this unusual air flow, both as to quantity and direction. The most important of these is the fact that the three train stops in this area are so arranged that a train entering this section from either direction, north or south, travels at a higher rate of speed than when leaving it. For a considerable distance beyond the end of the platform, the incoming train continues at relatively high speed. Consequently, large volumes of air are pushed and pulled from the tubes into the open platform area by each incoming train. The movement of air continues even after the train comes to rest because of the velocity pressure of the air. Because of the berthing arrangements, a departing train, however, stops for passengers just before it enters the parallel tube section. Consequently, it blocks the air movement and largely annuls the effect of velocity pressure. Furthermore, when a departing train leaves the continuous platform section, it does not accelerate rapidly enough to produce an outward air movement equal to the inward movement on the opposite track until its rear end has passed the first blast relief shaft. As a result of these train movements, air is being constantly pumped into this downtown area. The only place it can escape is through the various station passageways.

#### INVESTIGATION OF EFFECTIVENESS OF BLAST SHAFTS

As previously stated, blast shafts are connected to the train tubes beyond both ends of each of the station loading platforms. Their primary function is to relieve the piston action air blasts in stations, stairs and escalators and other public areas. These shafts, referred to as blast shafts, serve also as ventilating shafts. However, their close proximity to the stations, coupled with the fact that some of the outlying stations are colder in winter than the downtown stations, gave rise to the thought that unless it could be demonstrated that they actually accomplish a substantial reduction in air velocities throughout each station, it might be well to close the blast shafts during colder weather, and, in future subways, locate them farther from the station.

These possibilities were thoroughly investigated in a special study at the Chicago Avenue station. The arrangement of blast shafts at this station is typical of all outlying stations. In each of the two tubes, the approach and departure blast shaft areas total 200 sq ft and 100 sq ft, respectively, making an aggregate of 600 sq ft. These shafts were temporarily closed by placing heavy tarpaulins and planks over the sidewalk gratings. Air velocity readings were then taken for comparison with readings which had been taken before the shafts were closed. Simultaneously a record was made of train movements in both tubes. The observed results were plotted to show graphically

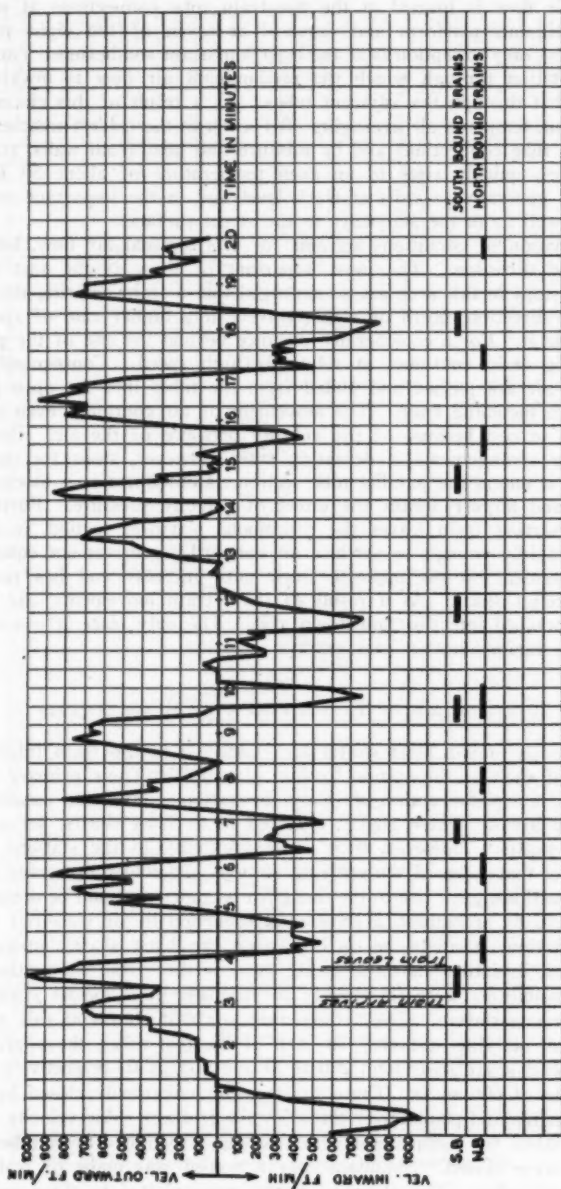


FIG. 6. AIR VELOCITIES IN CHICAGO AVENUE STATION WITH ELAST SHAFTS CLOSED

the effect of closing the blast shafts. Representative curves of *before and after* air velocities on escalators are shown in Figs. 5 and 6.

A comparison of these two curves brings out the following facts:

1. With the blast shafts closed, the reversals of direction and variations in magnitude of the air flow were far more numerous and violent than was the case when the blast shafts were open. Thus it is apparent that the closing of the blast shafts results in a great increase in the draftiness of a station.

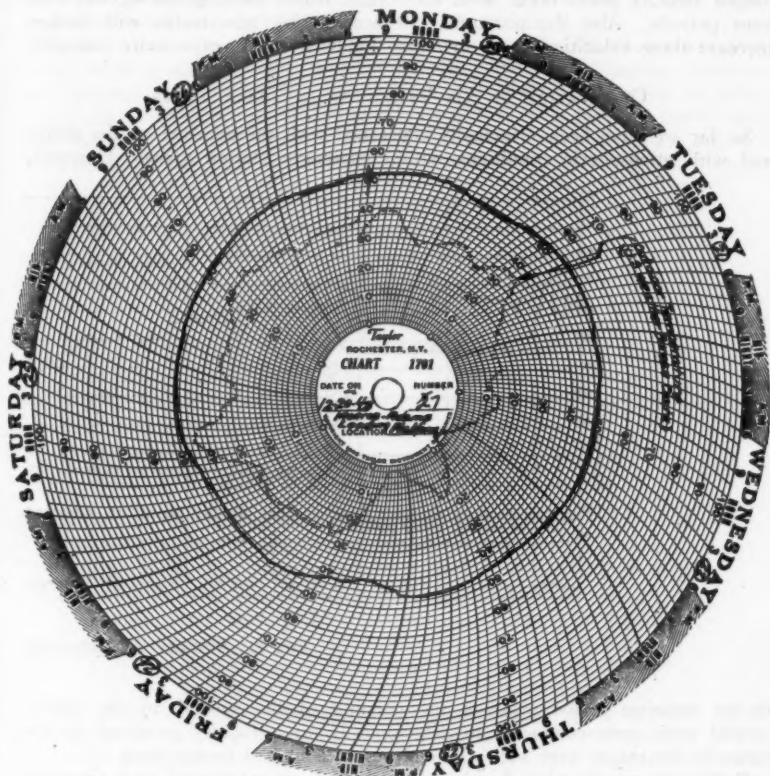


FIG. 7. TYPICAL TEMPERATURE RECORDER CHART

2. During a 20-min period when the blast shafts were closed, there were 14 different occasions, totaling  $5\frac{1}{2}$  min, when the air velocity exceeded 600 fpm. In a similar period with the blast shafts open, this velocity was exceeded only seven times for a total of  $1\frac{1}{2}$  min.

The tests at Chicago Avenue station, and similar tests at Van Buren-Congress station, demonstrate that the blast shafts are effective in reducing air velocities in stair and escalator wells and in mezzanines. The tests also

indicate that warming up of a station in winter cannot be accomplished by closing the blast shafts inasmuch as their closure would result in a very substantial increase in the amount of outside air that would be drawn into the subway by way of the mezzanine and stair and escalator wells. The net result would be a draftier and colder station.

Attention is called to the fact that these observations were made during a period when short trains were in operation and that higher and more prolonged velocity peaks occur with the longer trains running during the rush hour periods. Also the proposed fast accelerating new trains will further increase these velocities so as to make the blast shafts even more essential.

#### OBSERVATION OF AIR VELOCITIES IN TRAIN TUBES

So far we have been principally concerned with the comfort of the public and with atmospheric conditions within passenger station areas. However,

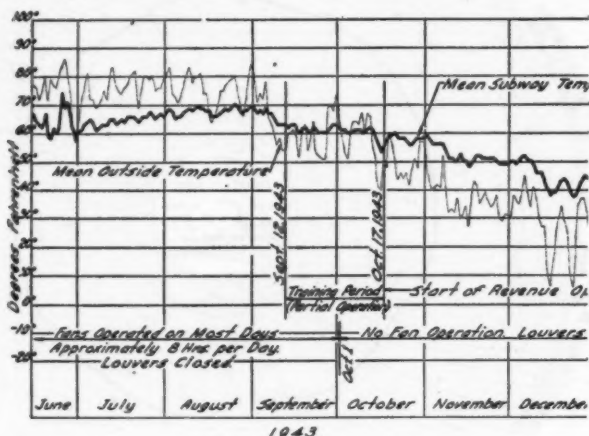


FIG. 8. SUBWAY TEMPERATURE

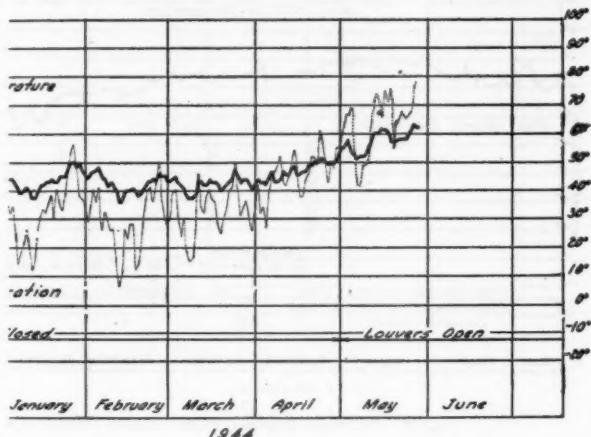
in the training period prior to the opening of the subway to the public, special tests were conducted to determine if air velocities produced in the tubes by the trains were such as to represent a hazard to trackmen.

During these tests, air velocities approximately as anticipated were observed on the emergency walk and the trackmen's walk. Peak velocities were roughly equal to one-half of the train speed; for instance with a train at 38 mph (3,344 fpm), the air flow gradually rose to 1,600 fpm opposite the front of a train. It dropped to near zero during the four seconds required for the train to pass, jumped to 1,700 fpm opposite the rear of the train and then very gradually subsided. Had the vent louvers been open, the flow produced by the train would have been increased so that the volume would have been considerably greater than the net piston displacement of the train, or approximately as had been calculated prior to construction.

The highest velocities were found to take place in the 4 ft-6 in. wide by

7 ft-6 in. high access openings in the dividing wall between tubes. These are spaced at intervals of about 200 ft. At train speeds over 30 mph (or 2,640 fpm) these air velocities went beyond the 2,500 fpm limit of the velometer and were estimated to be the equivalent of train speed. These peaks occur as each train passes and are followed by a violent reversal in direction of air flow. It was realized that trains on the northbound and southbound tracks would occasionally be timed so as to greatly increase the unbalanced pressure. Since workmen instinctively take refuge in these openings, grab irons were installed as a safety precaution.

At each of the three side platform stations, there are 16 similar openings in the dividing wall between the tubes for the purpose of balancing pressures and limiting air velocities in escalator and stair-wells. While not so effective in this respect as the open central platforms, they function better than anticipated. Peak velocities of 1,600 fpm through these openings are attained in spite of the fact that all trains come to a complete stop alongside of them.



CHART—MEAN FOR ENTIRE SUBWAY

Air velocities taken in the blast shafts adjacent to stations were found to vary widely at different points within the structure, particularly within the lower ports, and at the sidewalk grating. However, satisfactory metering conditions were found at a central location within the vertical shafts. With 4-car trains, either accelerating or decelerating because of the station stops, peak velocities in and out usually ranged between 500 and 700 fpm. While the average flow approached one-half of the maximum and continued in one direction or the other over 90 per cent of the time, the outward flow was very pronounced in the first shaft of a blast shaft unit. The last shaft in the group favored inward flow following passage of the train.

To date extensive meter readings in vent shafts have been deferred because the louvers were closed. Due to the higher train speeds opposite these vent shafts they should be more effective than the blast shafts. This is indicated by preliminary readings up to 1,000 fpm taken at the sidewalk level with the

vent shaft louvers open. Such peaks are of brief duration, and there have been no reported instances of nuisances to pedestrians. The survey of air movement in these various groups of shafts is being continued for data to be used in future design. It is now evident, however, that both the blast shafts and the intermediate vent shaft are functioning as planned and that they will be effective in providing ventilation and in reducing air velocities resulting from train movements.

#### SUBWAY TEMPERATURES AND ATMOSPHERIC CONDITIONS

In designing the ventilation equipment and structures for the Chicago subways, the principal considerations were adequacy of capacity for heat disposal in summer and the reduction of air velocities in public places. Each

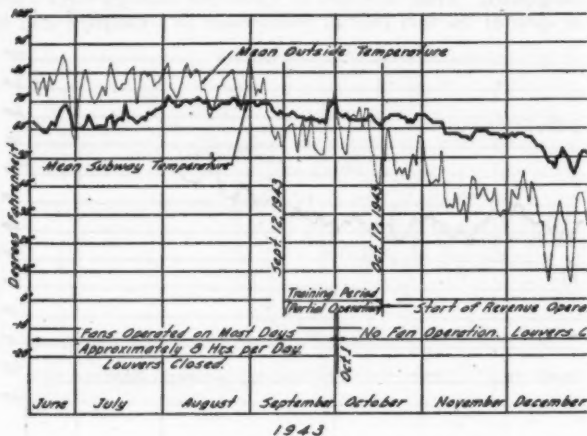


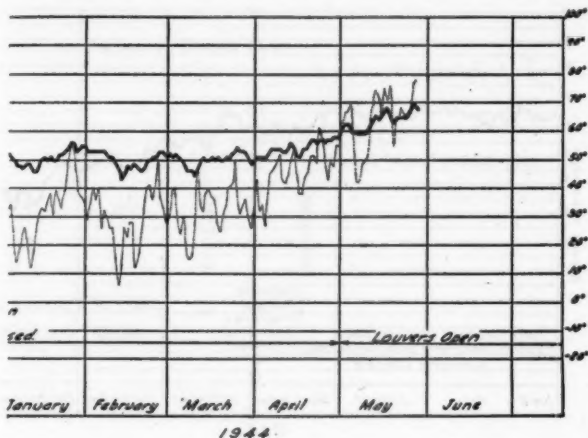
FIG. 9. SUBWAY TEMPERATURE CHART—MEAN FOR

peak hour, 297 cars now pass a given point in State Street subway. The total cars per day is 2,980, giving 14,900 car miles for the 5 miles of 2-track route. Electrical energy in kilowatt hours per car mile for train propulsion and car heating is estimated at 4.15 in winter and 3.45 in summer, with a weighted annual average of 3.85. The mean consumption of 57,365 kw hours, multiplied by the thermal conversion factor (3413), gives a daily heat input of 196 million Btu per day. Additional electricity for subway lighting, equipment motors and space heaters, including emergency direct current service, reached a winter maximum of 16,000 kw hours per day. It will average 12,000 kw hours, or the equivalent of 41,000,000 Btu per day. Approximately 100,000 passengers enter the subway stations daily and an equal number enter or pass through the subway on trains. This is equivalent to 50,000 man hours per day at an average of 15 min per trip. Assuming 400 Btu body heat per man hour, there is a further addition of 20 million Btu per day. It will be noted that the present combined heat input of 257 million Btu per day is originating with trains averaging not closer than

a 3 min headway. Facilities for the subway have been designed, however, for double the present traffic, and with trains operating on a 1½ min headway.

Little heat will permanently escape through the thick concrete walls and surrounding earth. The walls and earth act as a thermal accumulator of enormous capacity, absorbing heat in summer and returning it to the subway in winter with minor cycles of heat flow constantly taking place. The theoretical heat balance, previously mentioned,<sup>8</sup> included a determination of the probable production of heat, the accumulator effect of the concrete walls and the surrounding earth and a determination of the number of air changes required.

Fig. 7 is a thermometer record from the train platform at the Monroe-Adams station for a typical winter week. For purposes of comparison, hourly outdoor temperatures were also plotted on the chart. With outside



STATION PLATFORM AT MONROE-ADAMS STATION

sub-zero temperatures, the platform temperature dropped to 40 F. The usual winter temperature of this location was regular, however, and averaged 50 F. This approximates the mean annual Chicago temperature, as well as the temperature of the earth at platform depth. It is evident, then, that the concrete and the earth materially iron out wide fluctuations of temperature in the outside air drawn into the subway and are also most effective in leveling the temperature peaks which would otherwise occur during the periods of heavy traffic. This effect is somewhat less pronounced at locations where there is an air intake nearby because the cold air does not have time to become tempered by mixture and heat absorption from the surrounding walls and earth.

Fig. 8 is a chart of the mean daily temperatures for all train platform, mezzanine station and train tube locations in State Street subway at which recording thermometers are installed, with corresponding outdoor temperatures

<sup>8</sup> Loc. Cit., see Note 2.

plotted for comparison. By partial fan operation during the summer of 1943, when this subway was being finished, the average temperature in the subway was raised to 10 deg above that in Dearborn Street subway where construction had been deferred and fans were not ready for operation. The temperature in the latter subway was frequently below the atmospheric dew point. Periods of heavy condensation resulted. Poor air circulation retarded evaporation and caused these periods of condensation to overlap. In the State Street subway, as a result of fan operation, prevailing temperatures during the final stages of construction were kept above the dew point. On infrequent occasions when some condensation occurred, it quickly dried, and did not interfere with the installation of equipment and station finish.

Between September 12, 1943, and October 17, 1943, trains were run in the State Street subway for instruction of crew men and other operating

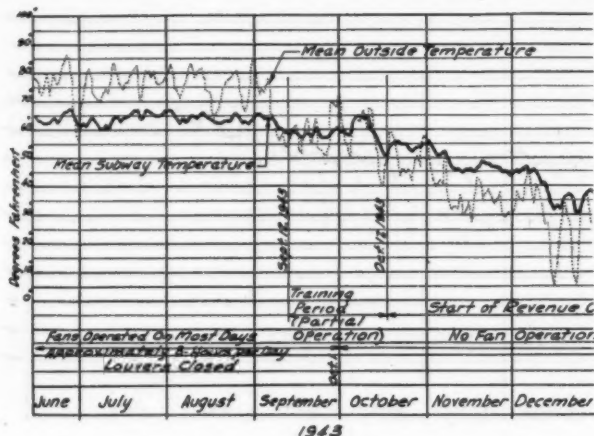


FIG. 10. SUBWAY TEMPERATURE CHART—MEAN FOR

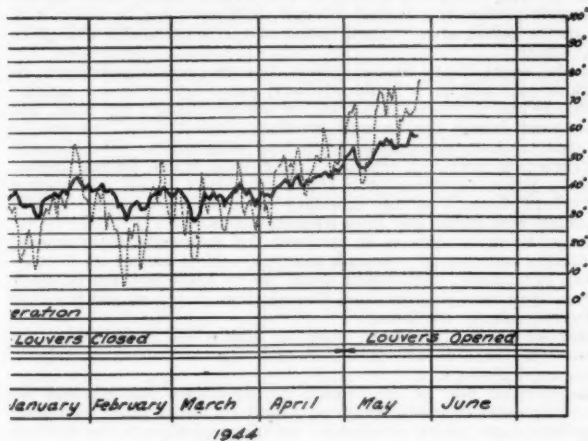
personnel. During this period both the subway temperature and the outdoor temperature averaged about 60 F, the partial train operation giving out sufficient heat to balance the cooling effect of the subway walls. For almost two months following October 17, 1943, at which time regular train operation started, a gradual decrease in outdoor temperatures was reflected in the subway by a more gradual decline to an average temperature of about 50 F.

Sub-zero temperatures in December were reflected by a pronounced drop in average subway temperature to the vicinity of 40 F, where it remained throughout the winter except for a rise to 50 F late in January, when unseasonably high outdoor temperatures prevailed. A comparatively balanced condition obtained in the subway throughout the winter, however, because the usual cold outdoor air was tempered to within 10 degrees of earth temperature by the heat given out by the walls and by the heat produced by car motors and other sources.

During spring months the subway will lag behind the rising outdoor temperatures and an occasional hot, humid day may cause the cooler subway

air to reach the dew point. To guarantee against condensation, subway temperatures will be raised by remote control of louvers and by operation of fans, if necessary, so as to draw in hot air when available. When the outdoor temperature drops sharply, the louvers will be closed and the fans will be shut off to reduce the intake of cold air from the outside.

While the accumulator capacity of the concrete and earth is very great, the rate of heat flow and absorption is insufficient to dispose of the great quantity of heat which will be produced in the Chicago subways. The excess heat must be carried out of the subway by frequent air changes. Because of the frequent air changes effected by piston action, and by fan operation when necessary, a differential of only a few degrees between subway and normal outdoor temperatures will be sufficient to remove the excess heat and maintain comfortable temperatures within the subway.



STATION PLATFORM AT HARRISON STREET STATION

Fig. 9 is a chart of mean daily temperatures at the Monroe-Adams station train platform. It is of interest in that it is typical of the uniform high winter temperatures prevailing in the important downtown stations due to the characteristic air flow previously discussed.

Fig. 10 is a similar chart for the Harrison Street station platform. It is selected because this station platform is the coldest in the State Street subway which is due to the fact that the prevailing flow of air in that vicinity is inward.

The predominant outward flow of tempered air through most of the station sidewalk entrances has been of value in preventing the accumulation of snow and ice on the steps. These entrances are not covered, but on three sides there is an open pipe railing mounted on a solid granite base 12 in. high. The protection afforded by the granite base and the outward currents of warm air were sufficient during the recent winter to prevent snow from drifting in, even with heavy snow falls and high winds. Whatever snow fell on the steps usually melted quickly so that during the winter there were

only a few occasions when a small amount of labor was required to free the outer steps of ice and snow.

#### CONCLUSIONS

Observations to date warrant the following conclusions:

1. Ventilation resulting from train piston action is providing the subway with fresh air far in excess of that required for breathing purposes, and, when supplemented by fan operation during hot periods, will be ample for heat removal even with ultimate traffic. This was the principal objective that was sought in designing the ventilation system.

2. Completion of the ventilation survey for the important downtown portion of State Street subway indicates a measured volume in excess of 400,000 cfm or the equivalent of 11 air changes per hour, due to train piston action within the 3,400 ft of route. All of this air does not come directly to this station section from out-of-doors since a considerable portion is admitted through the train tubes in which it is warmed on its way to the public areas. Such tempered air is preferable and gives increased comfort in winter.

3. The survey for the entire State Street subway has progressed sufficiently to indicate that, with trains operating on a 3-min headway, piston action is now supplying approximately 900,000 cfm of fresh outdoor air, or  $4\frac{1}{2}$  air changes per hour, and this rate will be increased during the warmer months to about 6 air changes by opening the remote controlled motor operated louvers at vent shafts. The Chicago subways were constructed to serve an ultimate traffic with trains having higher accelerating and braking power operating on a  $1\frac{1}{2}$  min headway. This will require greater ventilation for heat disposal but the necessary increase in air movement will automatically be produced with faster and more frequent trains.

4. Air velocities in public areas such as in escalator and stairwells, in mezzanine stations, at train platforms and through sidewalk gratings, seldom exceed an average of a few hundred feet per minute. Infrequent peaks up to 1,000 fpm are of short duration. No nuisances have been presented and the velocities do not exceed practical limits imposed by inherent subway conditions.

While air velocities in non-public places, such as within the train tubes, are relatively much greater no hazards are presented and they are approximately in accordance with design calculations.

5. Due to the characteristic flow of air toward the downtown stations and the predominating outward currents at street entrances, a uniform temperature averaging 50 F prevails in this important section in winter. At outlying stations the average winter temperature is approximately 10 deg lower.

6. The survey of ventilation and atmospheric conditions in State Street subway will be continued and extended to include Dearborn Street subway when the latter is placed in operation, the object being further to confirm conclusions so far drawn and to provide detail information to be used in future design.



**1263**

## DESCRIPTION AND PERFORMANCE OF TWO HEAT PUMP AIR CONDITIONING SYSTEMS (Using Well Water and Outside Air as the Heat Source)

By PHILIP SPORN\* AND E. R. AMBROSE,\*\* NEW YORK, N. Y.

SINCE the installation of the heat pump system at the Atlantic City Electric Company's office building, Salem, N. J.,<sup>1</sup> six other such systems have been installed on the properties of the subsidiaries of the American Gas and Electric Co.<sup>2</sup> The Ohio Power Company's office building systems at Portsmouth and Coshocton, Ohio, were selected for the subject of this paper, partly because of their unusual design, but mainly because the source of heat is air in one case and water in the other, thus offering an opportunity to compare the operation and coefficient of performance of two different heat source systems of the latest design which are located in approximately the same sections of the country.

### BUILDINGS

The Ohio Power Company's office building, Coshocton, Ohio, erected in 1940, has two stories and a basement, and is 88 ft long by 55 ft wide by 35 ft high.

The Ohio Power Company's office building, Portsmouth, Ohio, also erected in 1940, has four stories and is 104 ft long by 45 ft wide by 45½ ft high. Since this building is located in a flood area, no basement was provided. The fourth story was designed for equipment usually found in a basement and contains the air-conditioning room, transformer vault, telephone switching room and general storage vault.

Both buildings are of non-combustible construction with reinforced concrete foundation. Floor construction consists of reinforced concrete on open web joists and the roof consists of steel plate deck similarly supported. The space between the floor slab and the ceiling of the floor underneath was designed to afford the free movement of air and is used as an air return plenum for the heat pump systems. In general, all the interior walls and the ceilings above the first floor are plastered. The ceilings of the first story have acoustic tiles and all the floors are covered with asphalt tile.

All windows are fixed and have double glazing. The exterior walls for the Coshocton building consist of 4-in. face brick backed up by 8 in. of clay

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<sup>1</sup> An All Electric Heating, Cooling and Air Conditioning System (Installation and Performance of Plant Using the Heat Pump Principle), by Philip Sporn and D. W. McLenegan (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935.)

<sup>2</sup> The Heat Pump (An All-Electric Year-Round Air-Conditioning System), by Philip Sporn and E. R. Ambrose. (Heating and Ventilating, Vol. 41, No. 1, January, 1944.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Grand Rapids, Mich., June, 1944.



FIG. 1. OFFICE BUILDING OF THE OHIO POWER COMPANY,  
COSHOCTON, OHIO—WEST VIEW

terra-cotta blocks. In the Portsmouth building the exterior walls, up to the flood level, were made of solid masonry and lined with ceramic glazed block to make it water resistant. Above the flood level, 13 in. of brick was used with a plastered finish. One inch of rigid insulation was applied on the

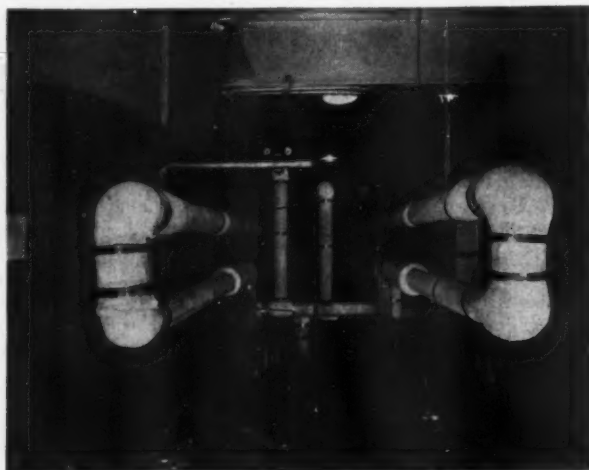


FIG. 2. REFRIGERANT COMPRESSORS AND CONNECTING PIPING,  
COSHOCTON SYSTEM

inside of all exterior walls in both buildings and 1 in. on the roofs at Portsmouth and 2 in. at Coshocton.

Fig. 1 is a west view of the Coshocton office building showing the main entrances. Fig. 2 is a view of the heat pump equipment room located in the basement of the Coshocton office building, showing the two compressors, some of the refrigerant and water piping, the 3-way valves, the conditioner housing and the pneumatic control panel. Fig. 3 is an east view of the



FIG. 3. OFFICE BUILDING OF THE OHIO POWER COMPANY, PORTSMOUTH, OHIO—EAST VIEW

Portsmouth building showing the main entrance. The second and third floors are divided into various individual offices.

Figs. 4 and 5 are views of the fourth floor air conditioning equipment room of the Portsmouth building showing the refrigerant compressors, control panel, air conditioner housing, outside air unit, condenser, cooler and connecting piping.

The liquid circulating pump is shown in the foreground of Fig. 5 and the "Freon" refrigerant system and liquid piping in the background. The electric control panel is at the left. The air conditioner housing and the pneumatic control panel are shown at the left in Fig. 4.

#### DESIGN TEMPERATURE LIMITS

The climatological data, as published by the U. S. Department of Agriculture, was studied for a five-year period to determine the minimum and maximum temperature which would be encountered during the heating and cooling seasons at the two locations. Based on this study,  $-5^{\circ}\text{F}$  and  $0^{\circ}\text{F}$  were selected as the probable average minimum outdoor temperatures which would



FIG. 4. AIR CONDITIONING EQUIPMENT ROOM, FOURTH FLOOR, PORTSMOUTH BUILDING

be encountered during the heating cycle at Coshocton and Portsmouth; respectively, and 95 F dry-bulb and 75 F wet-bulb as the average maximum temperature encountered during the cooling cycle. For both systems an inside temperature of 72 F and 30 per cent relative humidity was selected for the

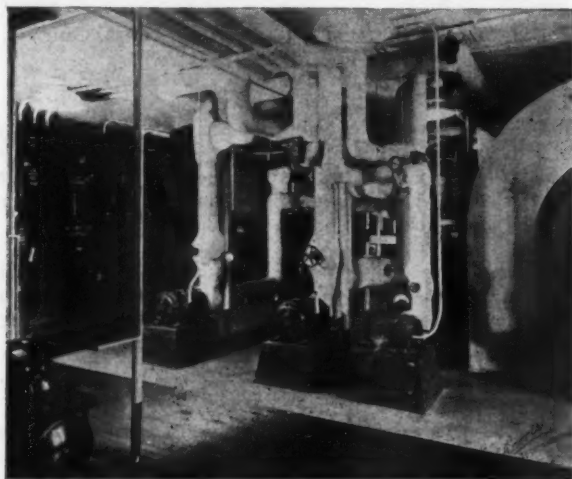


FIG. 5. AIR CONDITIONING EQUIPMENT ROOM, FOURTH FLOOR, PORTSMOUTH BUILDING

TABLE 1—HEAT GAIN—HEAT LOSS CALCULATIONS  
COSHOCOTON BUILDING

	HEAT GAIN BTU/HR AT 78 F-50 PER CENT R.H. INSIDE 95 F D.B.-75 F W.B. OUTSIDE	HEAT LOSS BTU/HR AT 72 F-30 PER CENT R.H. INSIDE -5 F OUTSIDE
Sensible:		
Conduction.....	26,600	206,700
Solar radiation.....	103,000	.....
Light.....	128,000	.....
Ventilation air.....	39,600	203,000
People.....	32,000	.....
Latent:		
Ventilation.....	38,000	.....
People.....	37,500	.....
TOTAL.....	404,700	409,700

heating season and 78 F and 50 per cent relative humidity during the cooling season.

Using the foregoing selected design temperatures, the heat gain and heat loss calculations for Coshocoton (Table 1) were about equal, figure 404,700 Btu per hours heat gain and 409,700 Btu per hour heat loss, while in the case of the Portsmouth Building (Table 2) the heat gain was 450,000 Btu per hour and the heat loss 354,700 Btu per hour.

After the heat gain and heat loss calculations are made, consideration can be given to the economical selection of the equipment. Usually the best method is to base the sizing of the compressors, coils and other equipment on the heat gain and using this selection determine the heat output at various outdoor temperatures. Following this procedure, the resulting heat output will be found to be sufficient to satisfy the heat loss for most installations. This was the case in Coshocoton, where the temperature of the heat source is constant and independent of the outdoor temperatures, thus allowing the refrigerating compressors to operate at approximately the same suction pres-

TABLE 2—HEAT GAIN—HEAT LOSS CALCULATIONS  
PORTSMOUTH BUILDING

	HEAT GAIN BTU/HR AT 78 F-50 PER CENT R.H. INSIDE 95 F D.B.-75 F W.B. OUTSIDE	HEAT LOSS BTU/HR AT 72 F-30 PER CENT R.H. INSIDE 0 F OUTSIDE
Sensible:		
Conduction.....	40,000	203,700
Solar radiation.....	82,500	.....
Light.....	181,500	.....
Ventilation air.....	31,800	151,000
People.....	44,200	.....
Latent:		
Ventilation air.....	30,400	.....
People.....	39,600	.....
TOTAL.....	450,000	354,700

sure during both the heating and cooling cycle. However, this is not the case in an air system such as Portsmouth. Here, as the outdoor temperature drops the suction pressure of the compressors is lowered, resulting in a proportional reduction in capacity. This necessitates, in most cases, an auxiliary heat source or some form of booster heaters during the lower outdoor temperatures.

In the case of the Coshocton system, it can be seen from the heat gain-heat loss tabulation that if the mechanical refrigeration equipment had been sized to remove the entire heat gain, the output during the heat cycle would have considerably exceeded the requirements for the coldest day. To remedy this condition, a well water precooling coil was utilized to absorb 120,000 Btu per hour, thereby reducing the mechanical refrigeration equipment and still satisfy the cooling requirement. Since the 10 ton reduction in the mechanical refrigerant equipment caused the heating capacity to be slightly undersized for the coldest days, a 15 kw electric heater was incorporated as a safety measure to act as a booster during these periods. It is interesting to note that in almost three years' operation of this plant no call has been made on this auxiliary heat source.

For the Portsmouth system, the equipment selected on the basis of heat gain requirements was found to have sufficient capacity to satisfy the heating requirement when the outdoor temperature was 20 F or above. For outdoor temperatures below 20 F city water is used as the heat source.

#### THE SYSTEM

The conditioned air is distributed to the several zones in each of the two buildings by means of galvanized iron ducts. The air is returned through grilles located in the outside walls near the floor (usually underneath the windows), which connect to the plenum between floor and ceiling by means of wall chases as shown in Fig. 6. These plenums open into a vertical shaft which in turn connects to the conditioner housing located in the equipment room.

A *blow through* type conditioning unit was employed on both the Coshocton and Portsmouth systems. In this design the air circulating fan delivers the outside-recirculated air mixture through the filters, then over the heating and/or cooling coils into one of two plenums. The zone thermostats, by controlling the operation of the two dampers, regulate the amount of air flowing from each of the two chambers into the zone supply duct.

The conditioning unit for the Coshocton building (Fig. 7) consists of a heating coil and humidifier in one plenum and a well water precooling coil and a mechanical refrigeration coil in the other. Due to the possibility of having a large crowd of people in the auditorium, an additional dehumidifying coil was installed in the supply duct to this zone to take care of the high latent heat load.

Fig. 8 shows the conditioning unit for the Portsmouth office building. The bypass coil in this unit serves as a supplementary heating or cooling coil, as required. The main difference between the two units is the use of a separate heating and cooling coil at Coshocton while one coil serves at Portsmouth.

The unit used to absorb the heat from the outdoor air at Portsmouth is shown diagrammatically in Fig. 9. It consists of a coil, automatic intake

FIG. 6. CROSS-SECTION THROUGH PORTSMOUTH BUILDING, SHOWING SUPPLY AND RETURN GRILLES

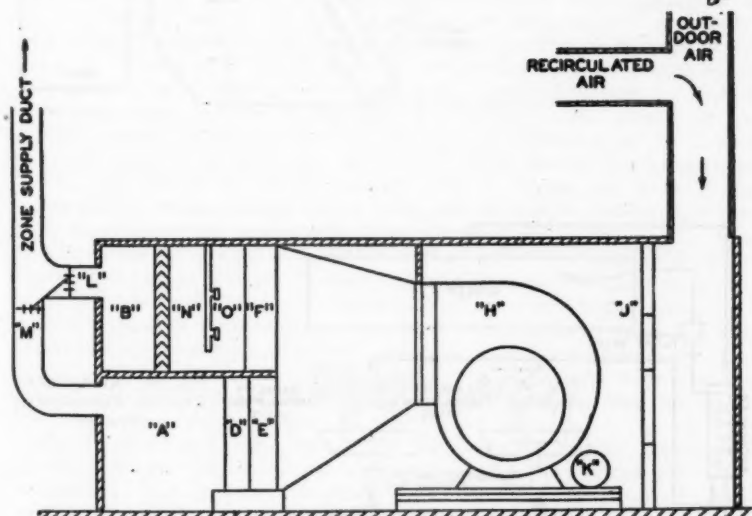
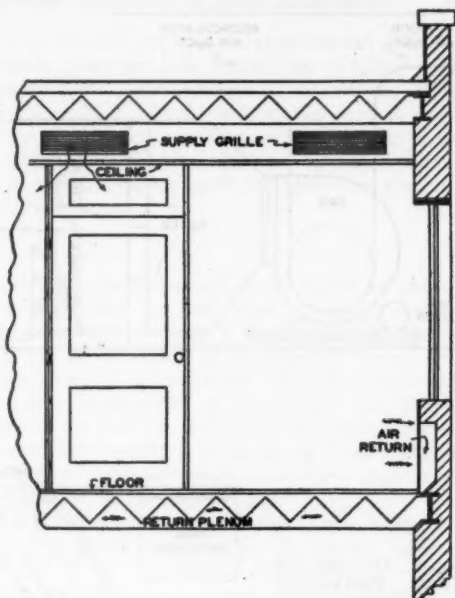


FIG. 7. CONDITIONING UNIT AT COSHOCTON

"A" Cold air plenum  
 "B" Warm air plenum  
 "D" Cooling coil  
 "E" Pre-cooling coil

"F" Heating coil  
 "H" Fan  
 "J" Filters  
 "K" Fan motor

"L" Damper  
 "M" Damper  
 "N" Eliminator plates  
 "O" Humidifying sprays

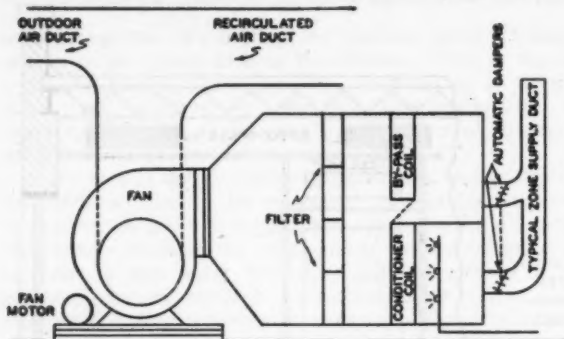


FIG. 8. CONDITION-  
ING UNIT AT  
PORTSMOUTH

FIG. 9. OUTDOOR AIR  
UNIT, PORTSMOUTH  
SYSTEM

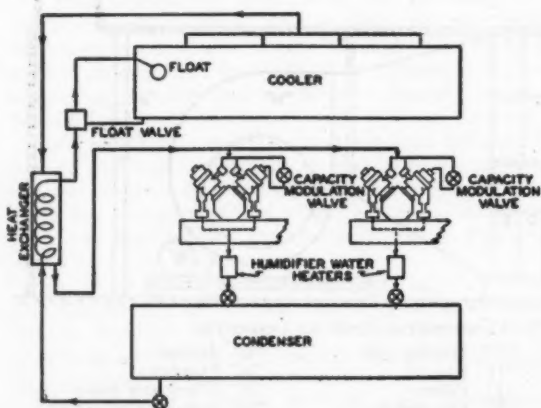
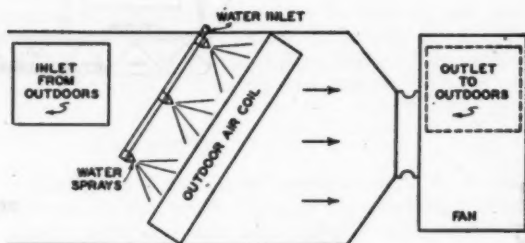


FIG. 10. REFRIGERANT  
CIRCUIT, PORTSMOUTH  
SYSTEM

and exhaust dampers, circulating fan and sprays. During normal operation the air is taken from the outside, over the coil, where heat is given up to the circulating medium, then discharged by the fan back to the outside. When the outdoor temperature falls below 20 F, the circulating fan is shut down, the intake and exhaust dampers closed, and a continuous flow of city water is sprayed over the coil, in which case all heat absorbed by the system is taken from the city water. During the cooling cycle the spray-coil combination is used as an evaporative condenser.

### PIPING

The designs at Coshocton and Portsmouth are commonly known as indirect systems, since a liquid is used to transfer the heat from an outside medium to the refrigerant. Water is used as this liquid at Coshocton because the temperature need never go below 32 F while at Portsmouth Super Pyro anti-freeze solution is employed since it is necessary that the temperature of the liquid be below the freezing temperature of water when the outdoor temperature is 32 F or below. In an indirect system the refrigerant circuit, shown by Fig. 10, for the Portsmouth system is fixed, going from the compressor to the condenser, through the heat exchanger, float valve and cooler, then through the heat exchanger back to the compressor.

In the Portsmouth system the humidifier water is heated by the refrigerant gas by means of a heat exchanger, while at Coshocton the heated water is taken directly from the system. The Portsmouth compressors are each equipped with 50 per cent modulation valves, shown diagrammatically in Fig. 10, making possible 25, 50, 75 or 100 per cent capacity modulation. This capacity modulation consists in allowing a portion of the refrigerant gas to go directly from the discharge to the suction side of the compressor, thus reducing the amount of refrigerant being circulated through the system. A solenoid valve installed in this bypass line controls the amount of gas which is passed through the circuit. The capacity modulation on the Coshocton system is provided by starting and stopping either one or both compressors.

When the systems are on the heating cycle, the bypass coil in the Portsmouth system and the cooling coil at Coshocton can furnish cooling simultaneously to some of the zones while heating is supplied to the others. Conversely, when the systems are on the cooling cycle, the bypass coil in the Portsmouth system and the heating coil at Coshocton can furnish heating while the other coils furnish the required cooling. This feature has proven quite important in these systems, since some of the occupied space has little or no outside exposure, therefore, practically requiring cooling the year 'round, and too during the heating season the auditorium, when in use, usually requires cooling because of the heat gain resulting from the assembly of a large group of people.

Fig. 11 shows the water circuit for the Coshocton system. During the heating cycle the deepwell pump A circulates well water through the circuit consisting of V-4, direction 1-2, water coolers 1 and 2, cooling coils D and G, mixing valve V-3 then through valve V-7, direction 1-2, to the discharge well. The pump B circulates water through closed circuit consisting of V-6, direction 1-2, condensers 1 and 2, heating coil F, valves V-1 and V-5, direction 1-2.

During the cooling cycle deepwell pump A circulates well water through the circuit consisting of V-4, direction 1-3, precooling coil E, condensers

1 and 2, heating coil, mixing valves V-1 and V-5, direction 1-3 to the discharge well. Pump B circulates the water through closed circuit consisting of V-6, direction 1-3, water coolers 1 and 2, coil D, and auditorium coil G, mixing valve V-3, direction 1-3, back to pump B.

Fig. 12 shows the circuit for the Portsmouth system. During the heating cycle pump A circuit consists of V-3, direction 1-3, conditioner coil, valve V-4, direction 3-1, and condenser back to pump A. Pump B circuit consists of V-1, direction 1-3, outdoor air coil, V-2 direction 3-1, and cooler back to pump B. Pump C circulates the fluid from the inlet of the outdoor air to pump B. Pump C circulates the fluid from the inlet of the outdoor air

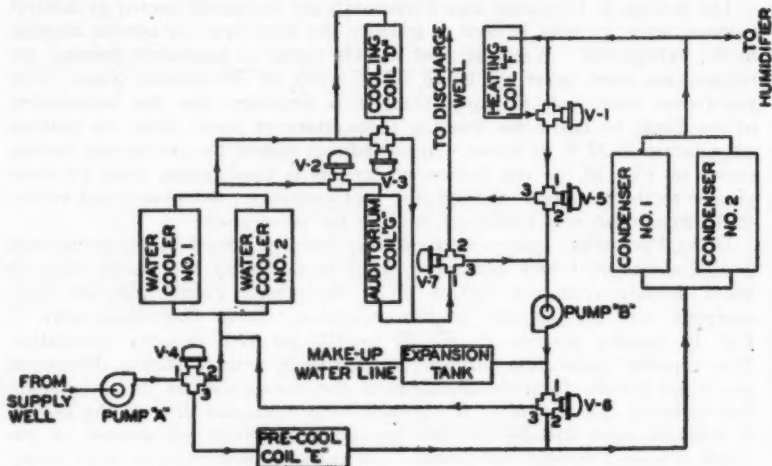


FIG. 11. WATER CIRCUIT, COSHOCTON SYSTEM, SHOWING CIRCULATING PUMPS AND REVERSING VALVES

HEATING CYCLE		COOLING CYCLE	
Valve V-4	Position 1-2	Valve V-4	Position 1-3
Valve V-5	Position 1-2	Valve V-5	Position 1-3
Valve V-6	Position 1-2	Valve V-6	Position 1-3
Valve V-7	Position 1-2	Valve V-7	Position 1-3

coil and/or the bypass coil, through V-5, then the bypass coil to the outlet of the outdoor air coil. This furnishes the means of cooling some of the interior zones when the remaining zones require heating. During the cooling cycle pump A circuit consists of V-3 direction 1-2, outdoor air coil, V-4 direction 2-1, and condenser. Pump B circuit consists of V-1 direction 1-2, air conditioner coil, V-2 direction 2-1, and cooler. Pump C circulates the fluid from the inlet of the outdoor air coil and/or the bypass coil, depending on the requirements of the thermostat, through V-5, bypass coil to the outlet of outdoor air coil. This allows heating of some of the exterior zones when the remaining zones require cooling.

Fig. 13 shows the city water circuit for outdoor air unit of the Portsmouth system. During the heating cycle the water to the sprays are controlled by valve V-9. This valve opens in response to outdoor thermostat T-3, shown

in Fig. 15. The water to the humidifier, under control of valve V-20, first passes through heaters which are located in the hot gas line from the refrigerating compressor. V-6 is open during this cycle, allowing all water in the drain pan to go to the sewer. During the cooling cycle pump D circulates the water from the drain pan to the sprays. During this cycle valve V-7 allows the make-up water to be under control of the float valve.

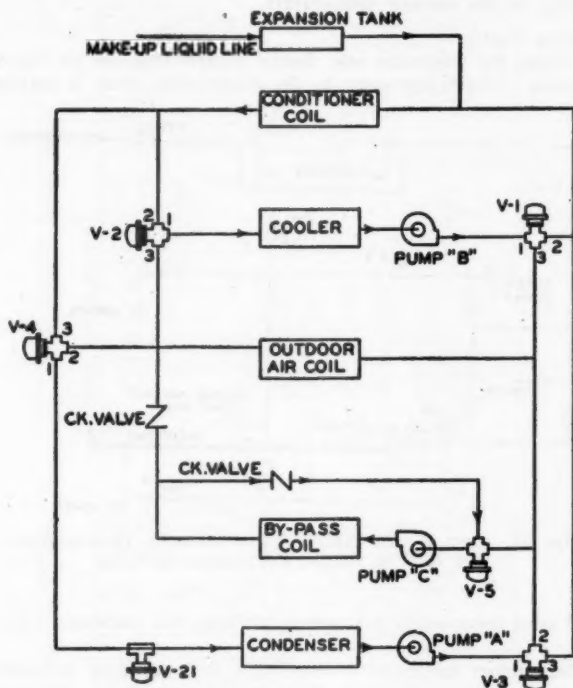


FIG. 12. SUPER PYRO CIRCUIT, PORTSMOUTH SYSTEM

Heating cycle—Valve position 1-3

Cooling cycle—Valve position 1-2

### CONTROL

Basically the two systems have similar control equipment. The reversing valves, shutoff valves, dampers and thermostats are pneumatically operated from a small motor driven air compressor system and are interlocked with the various electric starting switches controlling the motors. High and low pressure compressor safety switches and low temperature immersion thermostats, to guard against freezing of the cooler, have been incorporated in both designs. Also, all of the controls are interlocked with the fans so that the system is inoperative until the fans start. When the fan push button

is closed, starting the fans, the operation of both systems is completely automatic. The main difference in control of the two systems is the heating and cooling changeover feature. At Coshocton an outdoor thermostat automatically changes the system to the heating cycle when the outdoor air drops below a predetermined setting and to the cooling cycle when the outdoor temperature rises above the setting; while at Portsmouth two thermostats in the return air duct (set 5 or 6 deg apart) govern the cycle of operation independently of the outdoor temperature.

#### A—Coshocton System

Fig. 14 shows the pneumatic and electric control diagram for the operation of this system. The temperature in the conditioned space is maintained by

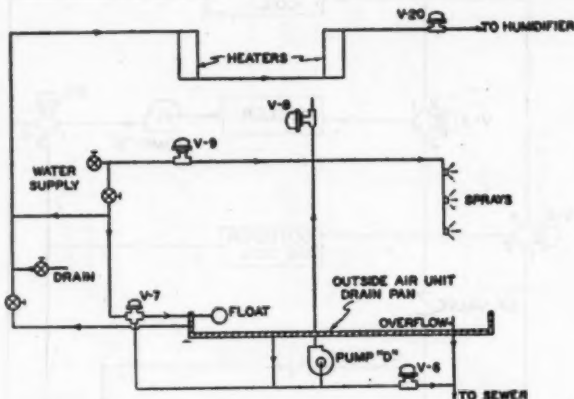


FIG. 13. CITY WATER CIRCUIT WHEN OUTDOOR TEMPERATURES ARE 20 F OF BELOW, PORTSMOUTH SYSTEM

the several zone thermostats T-1, which positions the automatic face dampers L and M.

When the outdoor temperature falls below the setting of thermostat T-11, all valves and controls change to their heating positions. The double-pole double-throw switch S-1 operates the 10 hp compressor through head pressure switch S-2; the immersion thermostat T-14, located in the condenser discharge, controls the 15 hp compressor through pressurestat P-4. Thermostat T-13, which controls the temperature in the plenum B by positioning water valve V-1, is readjusted by T-9 to a higher temperature setting as the outdoor air temperature falls. Pneumatic safety control thermostat T-15, located in the chiller discharge, and electrical immersion thermostats T<sub>B</sub> on the inlet and outlet of both coolers, stop the compressors if the water temperature falls below their setting. In case the compressors cannot maintain the desired temperature in plenum B, pneumatic electric switch P-1 will close the circuit to the 15 kw heater.

In order to provide the simplest control to automatically lower the night temperature during the heating cycle, an electric clock transfers the control from thermostats T-1 and T-13 to return duct thermostat T-8.

When the outdoor temperature rises above the setting of thermostat T-11, all valves and controls are changed to their cooling position. During this cycle switch S-1 operates the 10 hp compressor from the suction pressure switch S-3 and the immersion thermostat T-15 controls the operation of the 15 hp compressor. Return air thermostat T-12, which controls the air in the plenum A by positioning water valve V-3, is readjusted to a lower setting

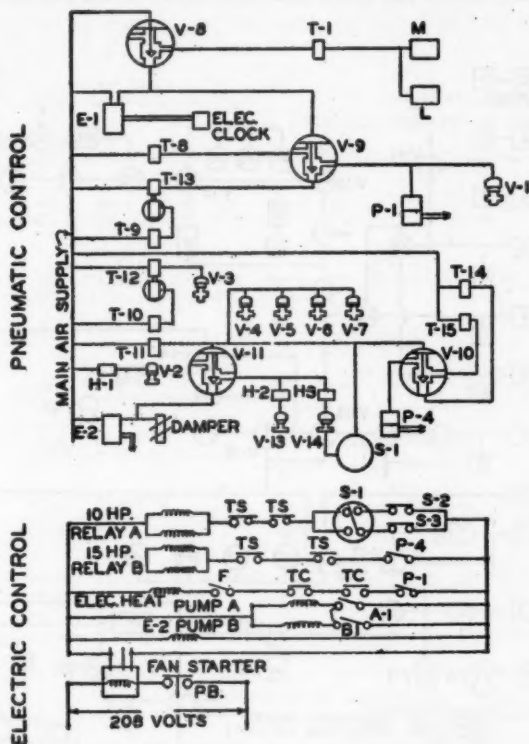


FIG. 14. ELECTRICAL AND PNEUMATIC CONTROL DIAGRAM, COSHOCTON SYSTEM

as the outdoor temperature rises. Auditorium humidistat H-1 controls valve V-2 on the auditorium dehumidifying coil G, opening the valve on a rise in humidity.

#### B—Portsmouth System

The pneumatic and electric control diagram for the operation of this system is shown by Figs. 15 and 16. The temperature in the conditioned space is maintained by zone thermostats T-4 to T-9, which modulate their respective face and by-pass dampers.

During both cycles of operation the compressors are electrically interlocked with the outside air fan so that one-half of the air flow is circulated when using one compressor and the full quantity when both compressors are operating. Pressurestats are installed on both sides of the cooler as a safeguard against freezing.

When the outdoor-recirculated air falls below the setting of the return airstat T-1, all controls and valves change to their heating position. Thermo-

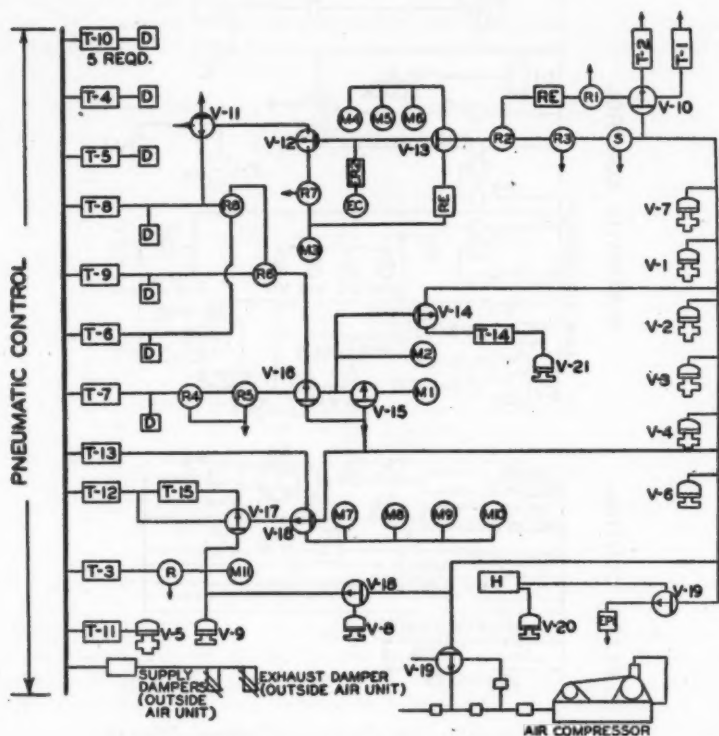


FIG. 15. PNEUMATIC CONTROL FOR PORTSMOUTH SYSTEM

stats T-6, T-8 and T-9, located in the zones which may require cooling when the other zones require heating, can operate pump "C" providing the face dampers on the bypass coil are fully open, in which case T-11 in the pump "C" discharge will control 3-way valve V-5 with modulation action to produce the desired liquid temperature going to bypass coil. When the outdoor air falls below 20 F, thermostat T-3 will stop the outside air fan, causing the intake and exhaust dampers to close and the city water supply valve V-9 to open, spraying water over the outside coil. The four steps of compressor

capacity during this cycle are under control of thermostat T-12, located in the liquid outlet of the conditioner coil.

During the night cycle the electric clock E, at a predetermined time, energizes magnetic pilot valve ER-2, thus transferring control of the heating cycle to thermostat T-8. The equipment will then operate intermittently in response to thermostat T-8.

When the outdoor-recirculated air rises above the setting of return airstart T-2, the system is changed to the cooling cycle. The four steps of com-

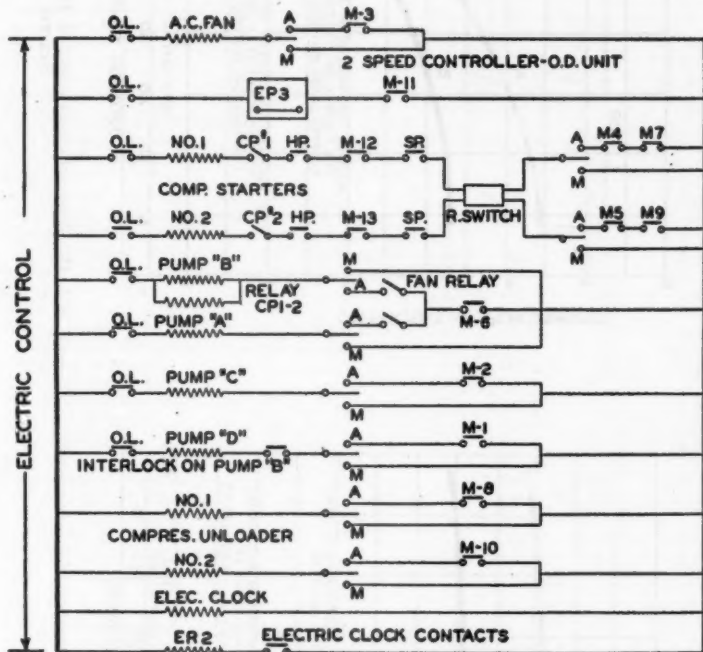


FIG. 16. ELECTRIC CONTROL FOR PORTSMOUTH SYSTEM

pressor capacity modulation are controlled by thermostat T-13, located in the liquid discharge from the conditioner coil. During this cycle the electric clock E, by energizing magnetic pilot valve ER-2, stops the fans and compressors through M4, M5 and M6 during the night period.

#### PERFORMANCE

Table 3 shows a sample of the test data obtained to check the coefficient of performance for the Coshocton system. These data indicate the probable temperatures of the circulating liquids in the two circuits and the resulting heat output of the system for three different operating cycles at substantially

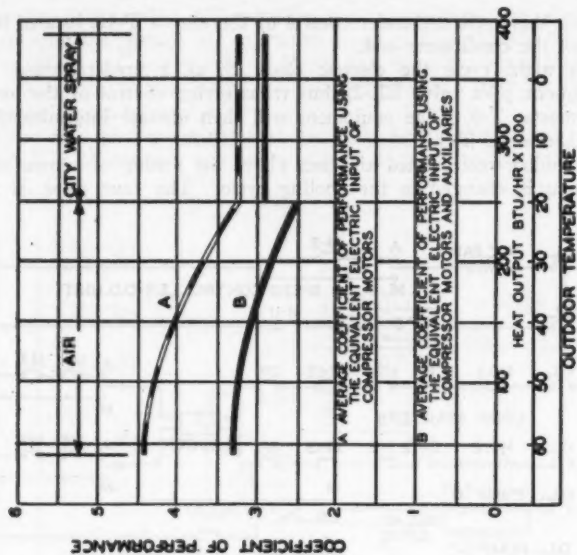


FIG. 18. AVERAGE COEFFICIENT OF PERFORMANCE, PORTSMOUTH SYSTEM

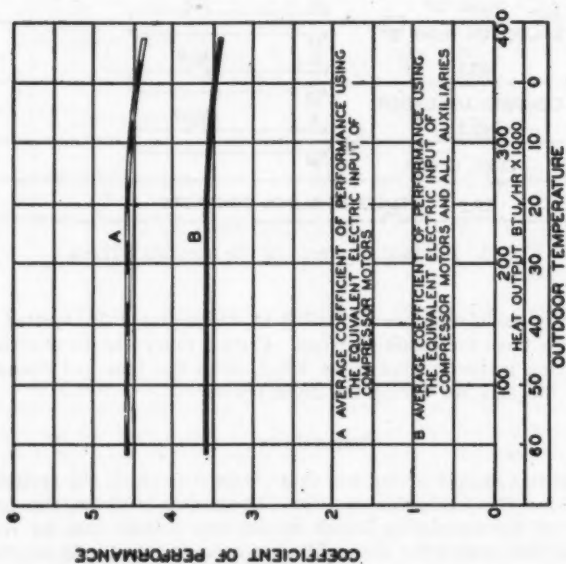


FIG. 17. AVERAGE COEFFICIENT OF PERFORMANCE, COSHOCTON SYSTEM

the same outdoor temperature. However, the data as they stand can be misleading since they do not represent a true balance for any particular operating condition of the system. It was necessary to make a series of such tests over the past two winters at various outdoor temperatures because it is practically impossible to obtain a period when the heating requirements of the building will exactly balance the output of the system. These data were then plotted and used together with manufacturer's data to obtain the

TABLE 3—ACTUAL PERFORMANCE DATA FOR COSHOCTON SYSTEM

	TESTS		
	No. 1	No. 2	No. 3
Supply Air to Conditioned Space			
Cubic feet per minute.....	11,000	11,000	11,000
Temperature, degrees Fahrenheit.....	78.6	80.5	94.3
Outside Air			
Cubic feet per minute.....	2,500	2,500	2,500
Temperature, degrees Fahrenheit.....	37	35	35
Conditioner Heating Coil (for location coil F see Fig. 11)			
Entering Water Temperature, degrees Fahrenheit....	88	92.8	113.5
Leaving Water Temperature, degrees Fahrenheit....	83	87.0	103
Gallons per minute.....	70	70	70
Water Cooler (see Fig. 11):			
Entering Water Temperature, degrees Fahrenheit....	55	55	55
Leaving Water Temperature, degrees Fahrenheit....	47	46.5	47.3
Gallons per minute.....	37	38	75
Electric Consumption:			
Compressors, kilowatts.....	10.2	15.5	27.5
Auxiliaries, kilowatts.....	5.2	5.2	5.2
Total kilowatts <sup>a</sup> .....	15.4	20.7	32.7
Capacity:			
Refrigeration, Btu/hour.....	148,000	161,500	288,600
Heat Output, Btu/hour.....	175,000	203,800	366,600
Coefficient of Performance:			
Using Kilowatt Input to Compressors.....	5.0	3.85	3.9
Using Total Kilowatt Input <sup>b</sup> .....	3.5	3.0	3.3

## Test No.

- 1— 10 Hp Compressor—120 psi head pressure—40½ psi suction
- 2— 15 Hp Compressor—140 psi head pressure—35 psi suction
- 3— 10 Hp Compressor—150 psi head pressure—42½ psi suction
- 15 Hp Compressor—175 psi head pressure—38½ psi suction

<sup>a</sup> Kw to conditioner supply fan not included.

<sup>b</sup> 50 per cent of auxiliaries kw input was considered useful heat.

curves shown in Fig. 17, giving weight to the fact that in many cases the compressor head pressure and the liquid circulating temperature were either higher or lower than would be experienced under balanced operating conditions. The solid lines indicate the average coefficient of performance using the electric input to the compressor motors and all the auxiliary equipment, while the broken line includes the compressor motor input only. A comparison of curves A and B shows that the auxiliary equipment reduces the coefficient of performance approximately 22 per cent.

The slight drop in the coefficient of performance for the Coshocton system, under maximum output, is partly due to the rising of the heating coil inlet

water temperature as the outdoor temperature falls and partly because the 10 and 15 hp condensers are the same size. This causes the 15 hp compressor to operate at a relatively high head pressure when both units are under full load, thus resulting in an increase in its power input and a corresponding reduction in the coefficient of performance.

TABLE 4—ACTUAL PERFORMANCE DATA FOR PORTSMOUTH SYSTEM

	TESTS			
	No. 1	No. 2	No. 3	No. 4
Supply Air to Conditioned Space:				
Cubic feet per minute.....	18,000	18,000	18,000	18,000
Temperature, degrees Fahrenheit.....	77.1	85.1	78.9	87.2
Outside Air:				
Cubic feet per minute.....	2,000	2,000	2,000	2,000
Temperature, degrees Fahrenheit.....	32	32	32	32
Conditioner Heating Coil (for location see Figs. 8 and 12):				
Entering Liquid Temperature, degrees Fahrenheit.....	85	97.2	87.5	100
Leaving Liquid Temperature, degrees Fahrenheit.....	81.0	90	83	92
Gallons per minute.....	95	95	95	95
Liquid Cooler (for location see Figs. 10 and 12):				
Entering Liquid Temperature, degrees Fahrenheit.....	21.2	19	28.5	26
Leaving Liquid Temperature, degrees Fahrenheit.....	18.2	13.8	25	20
Gallons per minute.....	95	95	95	95
Electric Consumption:				
Compressors, kilowatts.....	16.6	32.4	17.8	35.1
Auxiliaries, kilowatts.....	9.2	11.1	4.8	4.8
Total kilowatts <sup>a</sup> .....	25.8	43.5	22.6	39.9
Capacity:				
Refrigeration, Btu/hour.....	140,000	250,000	168,000	284,000
Heat Output, Btu/hour.....	187,200	342,000	218,500	383,500
Coefficient of Performance:				
Using Kilowatt Input to Compressor.....	3.3	3.1	3.6	3.2
Using Total Kilowatt Input <sup>b</sup> .....	2.2	2.4	3.0	2.9

## Test No.

- 1—1-25 Hp Compressor: 135 psi head pressure—11 psi suction
- 2—2-25 Hp Compressor: 160 psi head pressure—10 psi suction
- 3—1-25 Hp Compressor using water spray—140 psi head pressure—16 psi suction
- 4—2-25 Hp Compressor using water spray—170 psi head pressure—12 psi suction

<sup>a</sup> Kw to conditioner supply fan not included.<sup>b</sup> 60 per cent of kw input to circulating pumps included as useful work.

Table 4 shows a sample of the test data obtained over the past two winters to check the coefficient of performance for the Portsmouth system. The data were tabulated and plotted to give curves shown in Fig. 18. The same comments made on the Coshocton tests apply similarly to the Portsmouth tests. It will be noted that the coefficient of performance drops rather uniformly as the outdoor temperature falls until 20 F is reached, at which point water is introduced as the source of heat. The falling off of the

coefficient is caused mainly by the decrease in suction pressure resulting from the frosting of the outdoor coil and the temperature reduction of the heat source. Both of these phenomena cause the head pressure-suction pressure differential to increase and the Btu per kilowatt to decrease.

Defrosting of the outdoor coil at Portsmouth is accomplished by reversing the cycle of operation and/or spraying with water, but neither method is too satisfactory. Other defrosting methods are being investigated, but to date this remains a problem to be solved. However, the defrosting problem which has been experienced at Portsmouth cannot be considered a serious objection, since the difficulty occurs at rather infrequent intervals. On the whole, the system is operating quite satisfactorily without an undue amount of maintenance.

### CONCLUSIONS

The results of operation observation of, and tests on, the heat pump installations at Coshocton and Portsmouth lead to the following conclusions:

1. The heating system using water as the heat source operated satisfactorily throughout the range of outdoor temperatures encountered and the system employing air as the source of heat is satisfactory up to outdoor air temperatures of approximately 20 F.

2. The idea of using water as an auxiliary heat source, as carried out at Portsmouth, when outdoor temperatures are below 20 F, has not proven fully satisfactory mainly because the temperature of water available was around 38 F. This not only required a great quantity of water to obtain the necessary heat but also caused the compressor to operate at a relatively low suction pressure.

3. Operation of the air system at temperatures below 20 F has not proven satisfactory. On several occasions when this was attempted at Portsmouth, difficulty resulted because the frosting of the outdoor coil seriously affected the capacity.

4. Improvements in means and methods of defrosting the outdoor coil are a much needed development for successful heat pump installations using air as the heat source where temperatures below 20 F are encountered.

5. Experience both at Coshocton and Portsmouth indicates not only the desirability and need, but also the possibility of simplifying controls. Thus, at Coshocton there was shown on the whole little need for outdoor compensators to change the setting of the controlling thermostats. At Portsmouth justification has not been shown for the multiplicity of thermostats installed to obtain refinements of operation.

6. The indirect systems at Coshocton and Portsmouth have given less operation and maintenance trouble than the direct systems which have been installed elsewhere on properties of American Gas and Electric Co. It is believed that at least part of this improvement is due to having a fixed refrigerant circuit, thus eliminating probable trouble from reversing valves in the refrigerant circuit and erratic oil return. But most of the improvement is believed due to the advancements which have been made in the manufacture and installation technique of the refrigeration equipment. Further study and additional experience is necessary to determine which type of design is more economical and what improvements can be made in each system to achieve better year-round performance.

7. Average coefficients of performance of 3.0 on the air system and 3.6 on the water system are impressive results. Both are, however, susceptible to improvement as advancements are made in design and performance of the refrigeration compressor, heat transfer surfaces and other equipment making up the system which will result in increased operating suction pressures and in decreased head pressures. Also, considerable possibilities exist for reducing and in many cases completely eliminating the auxiliaries which in the case of Portsmouth and Coshocton systems reduce the average coefficient of performance approximately 22 per cent.

8. The heat pump, properly engineered and installed from an application and design standpoint, is even now a wholly practical device to furnish year-round all-electric air conditioning service. Its further development should make it even more so.

## ACKNOWLEDGMENT

The authors wish to acknowledge the cooperation and assistance of the engineers of the General Electric Co. and the Westinghouse Electric and Manufacturing Co., and William H. Torges, R. H. Anders and E. L. Vogt of The Ohio Power Co. for their help in making the observations and tests and following the operation of the systems.

## DISCUSSION

L. T. AVERY, Cleveland, Ohio: I would like to express my personal thanks and those of us who are members of the Society who are interested in air conditioning, to the authors for this fine paper. As you notice, it is the only one at this session that has any bearing on the summer aspects of air conditioning. I do not think that we can overlook the importance of the summer phases of this business. In fact, they should be emphasized. I recall that in January, in the panel discussion on the future trends in heating, one of our speakers brought out that the big demand for heating equipment of the future would be in the range of 30,000 to 40,000 Btu units for small homes. I suggested at that time that the refrigerating machine, the small three-ton and five-ton units, fitted right into that niche very nicely, and that we could produce the heating in the winter with these small efficiently insulated homes with the same machine that we cool them in the summer.

The difficulty is to find a free source of heat for the refrigerating machine to work against in the winter, and unless cold water from a free source or a well is available it looks as if the consideration in homes would be whether or not the person wants to go into the cold storage refrigeration business.

That line of thought did prompt, however, the idea that there are many places that do use refrigerating machines in the winter time. Skating rinks, for example, cold storage plants, dairies, ice cream plants, and in those plants the reverse cycle method of heating is an economically sound one, and because they do have a continuing need for probably more refrigeration than the need for the heat. So the heating and ventilating engineer is now over into the field of the refrigerating engineer and he must tie in there, as we found this morning he had to tie in with the industrial hygienist. He must be a *master* of all trades.

We have learned from the locker plant business the possibility that the extended use of food freezing in the home, will not be just a small unit but a fairly large unit. This might give us a use of refrigeration for heat, particularly in larger homes, clubs and small hotels.

I think we can say that in regard to the subject this paper is only suggestive of the thinking which will go far beyond the application here, where the utility is doing a one-purpose job and that is to waste the refrigeration in the winter-time in order to get the heat. I think there are many applications where the refrigeration can be used to obtain a much better economic solution than by discarding the refrigeration.

## In Memoriam 1944

NAME	JOINED
EDSON G. ARMOUR (Flying Officer, R.C.A.F., killed in action March 18)	1940
FREDERIC F. BAHNSON	1917
JAMES W. BASSETT	1938
BERNHARD P. BREDESEN	1931
PERCY J. BRYANT	1915
JOHN D. CASSELL ( <i>Life Member</i> )	1913
MERRIMAN C. GILLET	1916
ARTHUR L. HALLER	1930
MARRIOTT T. JOHNSTON	1939
CARL M. H. KAERCHER	1937
JAMES I. KRUEGER ( <i>Life Member</i> )	1921
ROY A. NORMAN	1937
CECIL N. PALMER	1943
ANDREW H. RAVEN	1937
MILTON L. ROSAS (Lieut. on Liberator Bomber, killed in action June 20)	1941
LESTER SEELIG	1925
HOWARD I. SCHULZ	1916
LAURENCE T. SHERWOOD	1937
W. E. STARK	1928
HOWARD M. WYLIE	1917



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